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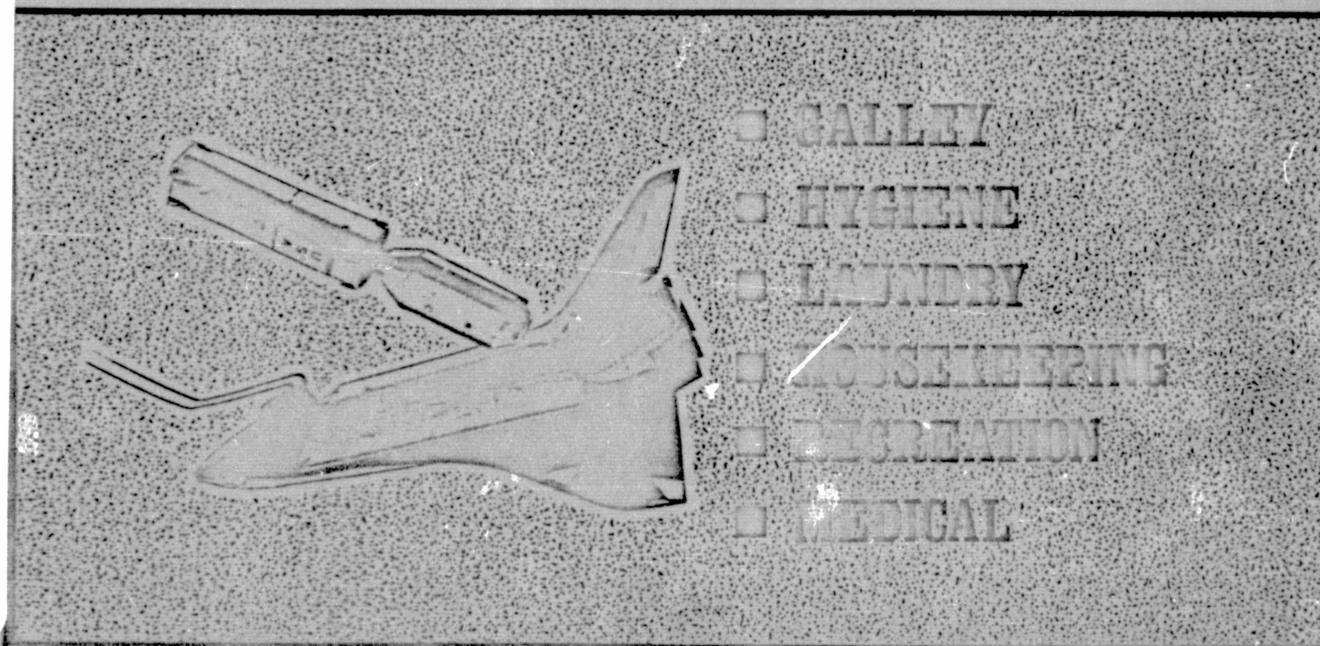
CREW APPLIANCE
COMPUTER PROGRAM MANUAL

CREW APPLIANCE STUDY



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CREW APPLIANCE
COMPUTER PROGRAM MANUAL

Contract NAS 9-13965

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Prepared for

National Aeronautics and Space Administration
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REVISIONS

REV. SYM	DESCRIPTION	DATE	APPROVED
A	Refinement of Space Station shower loop solution method viii F-5 G-9 6-9.1* F-9 G-31 6-22 F-11 6-23 F-15 6-24 F-20 through F-33 A-22 F-33.1 F-33.2 *Page 6-9.1 was inserted to explain the new shower solution method. Pages F-33.1 and F-33.2 were added to accommodate the larger GPOLY subroutine listings required with the new method. Simpler method used to compute saturated humidity conditions in SHOWER and WASDRY subroutines. 3-100 A-19 A-27 3-103 A-20 A-31 3-129 A-24 A-32 3-132 A-26 Initial water contained in Space Station clothes washer and dish-washer accumulators changed 6-16 6-32 G-13 6-17 F-6 6-29 F-7 Space Station potable water holding tank capacity revised. F-13 Computer print-out of final results from revised Space Station/ appliances system simulation. G-37 through G-64 Insert Revision Page Vol. 2 Ai Change Page Number Ai to Ai.1 Ai.1	10-16-75	C. L. Toliver

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ABSTRACT

Trade studies of numerous appliance concepts for advanced spacecraft galley, personal hygiene, housekeeping, and other areas were made by the Boeing Aerospace Company, Contract NAS 9-13965, to determine which best satisfy the Space Shuttle Orbiter and Modular Space Station mission requirements. In conjunction with these studies, analytical models of selected appliance concepts not currently included in the G-189A Generalized Environmental/Thermal Control and Life Support Systems (ETCLSS) Computer Program subroutine library were developed. This document describes the new appliance subroutines with complete analytical model descriptions, solution methods, user's input instructions, and validation run results. The appliance components modeled were integrated with G-189A ETCLSS models for Shuttle Orbiter and Modular Space Station, and results from computer runs of these systems are presented.

KEY WORDS

Appliance
Clothes washer
Dishwasher
Food management
Fortran
Mathematical model
Modular Space Station
Nodal network
Reverse osmosis
Shower
Shuttle Orbiter
Subroutine
Waste collection

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1.0 SUMMARY

Trade studies of numerous spacecraft appliances (Reference 1) have identified the most promising appliance concepts for the Space Shuttle Orbiter and Modular Space Station. Future spacecraft analysis effort will require simulation of these appliances using the G-189A ETCLSS Computer Program (Reference 2). G-189A subroutines are available for simulation of some appliance concepts, while others are so elementary as to require no special subroutine. Subroutines which model the remainder of optimum appliances chosen from the trade studies have been developed and are described in this report. The following six new subroutines were written, some of which model more than one appliance component:

- CHILLR - simulates a thermally insulated refrigerator or freezer locker cooled either by an externally chilled fluid or a self-contained refrigeration unit
- FTRAY - food warming/serving tray (Skylab-type)
- ROSMOS - reverse osmosis waste water treatment unit
- SHOWER - spacecraft whole body shower
- WASDRY - simulates a dishes or clothes washer, dryer, or washer/dryer combination. Will also model a towel/cloth drying rack.
- WASTEC - dryjohn (commode) and/or urinal

Complete user's input instructions for each subroutine are included in Section 3, together with the analytical model description and method of solution. The order of presentation of data for each subroutine corresponds to the format prescribed for new G-189A component subroutines in Reference 2. In Section 4 are presented performance

1.0 (Continued)

data for each appliance component model with constant dummy inlet and ambient conditions. These data are compared with test data where available, or with analytical predictions, to verify the accuracy of the subroutine models.

The appliance components were incorporated into G-189A models of the Space Shuttle Orbiter and Modular Space Station ECLSS to verify their performance and operational status within the G-189A software environment. For the Shuttle case described in Section 5, an available steady-state G-189A Shuttle ECLSS model (Reference 3) was used to simulate four orbital phases of operation. Results are presented for the basic unmodified Shuttle case, and for the same case except with the appliance models added. The computer results demonstrate that the appliance subroutines used are compatible with the G-189A program and accurately model the respective appliance components. No operational Modular Space Station G-189A ECLSS model is currently available, so a simplified model was developed and the optimum appliances selected from Reference 1 included. For this case a transient 10-hour simulation using typical appliance usage schedules, described in Section 6, was run. Complete results are plotted for the system as a whole and for each individual appliance component. Again, the results demonstrate the usage and accuracy of the new appliance subroutines.

This document, in its entirety, satisfies the requirements of contract DRL T-968, Line Item No. 5, "Digital Computer Program Requirements." A computer program "Users Manual", described in DRL Line Item No. 3, was combined into this single document due to the duplication of a large amount of data. The Users Manual consists of three basic parts. The first part--mathematical models, solution methods, and users input instructions--comprises Section 3 of this document. The second part--subroutine listings--comprises Appendix A. The third part is sample problem descriptions. These are given in Section 4, for the individual appliance components, and in

1.0 (Continued)

Sections 5 and 6 for the all-up appliance subsystems within a Shuttle and Space Station ECLSS. Additional data input and computer results for the Shuttle and Space Station systems are also given in Appendixes D through G.

2.0 INTRODUCTION

Trade studies of numerous spacecraft appliances (Reference 1) have identified the appliance concepts which best satisfy the Space Shuttle Orbiter and Modular Space Station mission requirements. Future spacecraft analysis effort will require simulation of these appliances using the G-189A ETCLSS Computer Program. This program, Reference 2, provides system level ECLSS performance simulation by performing mass and energy balances throughout all the interactive components and flow loops comprising a total system. Use of the G-189A program requires a subroutine for each component in the system. These subroutines are all similar in that a standard format of G-189A flow and thermodynamic data form the input for each. The input data for a given component are taken from the output data from the upstream component. The subroutine then must modify these input data in a manner which reflects the performance of the component it models, and present the output data in the required G-189A format. The program allows the user to control or modify the solution as it progresses by calling two subroutines, GPOLY1 and GPOLY2, immediately prior to and following each component solution. These routines may be used, for example, to alter fluid flow paths, turn components on or off, reevaluate component model data based on the current solution results, and compute and store parameters for later plotting.

The optimum appliance concepts selected from the trade studies in Reference 1 are shown in Tables 2-1 and 2-2 for Shuttle Orbiter and Modular Space Station. Some of these concepts do not require a new G-189A subroutine since (1) a routine is already available, (2) no thermal/mass exchange is involved, or (3) operation of the component is so simple it requires only a minor addition to the GPOLY routine logic. Appliances in this category are as follows:

o Reusable dishes, wet and dry wipes	None needed
o Vomitus collection	None needed
o Partial body washing, wet wipes	Simple GPOLY logic only required

2.0 (Continued)

o Partial body drying, dry wipes or electric dryer	None needed (or a simple heater using G-189A routine ALTCOM)
o Wet shave	GPOLY logic required for water usage only
o Windup razor	None needed (or a simple heater using G-189A routine ALTCOM if electric)
o Toothbrush	GPOLY logic required for water usage only
o Vacuum refuse collection	GPOLY logic only required, or G-189A routine ALTCOM for an electric heater
o Tape recorder, TV	GPOLY logic only required, or G-189A routine ALTCOM for an electric heater

For the remaining appliances, six new G-189A subroutines have been written, some of which will model more than one type of appliance. These subroutines have been designated as G-189A component subroutines number 66 through 71, and are generally described as follows:

Subroutine Number	Subroutine Name	Description
71	CHILLR	(simulates a thermally insulated locker cooled either by an externally chilled fluid or a self-contained refrigeration unit) * Freezer * Refrigerator
66	FTRAY	* Food warming/serving tray (Skylab-type)
69	ROSMOS	* Reverse osmosis waste water treatment unit
67	SHOWER	* Spacecraft whole body shower

2.0 (Continued)

Subroutine Number	Subroutine Name	Description
70	WASDRY	* Clothes washer * Clothes dryer * Combined clothes washer/dryer * Dishwasher/dryer * Towel/cloth drying rack
68	WASTEC	* Dryjohn * Urinal

A listing of the subroutines is given in Appendix A. They have been written in conventional Fortran V language and are operational on the NASA JSC SRU 1108 EXEC II computer system. This document presents complete user's input instructions, results of verification runs, and demonstration of their operation in all-up Shuttle Orbiter and Modular Space Station ECLSS simulation runs. The English system of units is a built-in feature of the G-189A program, both for input and output data. Therefore, all input data herein described were necessarily presented in English units to be compatible with the program. Output data are also presented exactly as generated by the program; i.e., in English units. Where specific test data are compared with analytical results, metric units are also included.

TABLE 2-1

SHUTTLE APPLIANCE CONCEPTS TRADE STUDY RESULTS

HABITABILITY SUBSYSTEM	HABITABILITY FUNCTION	APPLIANCE FUNCTION	CONCEPT CHOSEN	FIRST RATED CONCEPT	SECOND RATED CONCEPT
FOOD MANAGEMENT	FOOD STORAGE	REFRIGERATED	Space Radiator	Space Radiator	Thermoelectric
	FOOD PREPARATION	WARMING	Heating Trays	Heating Trays	Convective Oven
	GALLEY CLEANUP	DISH CLEANUP	Reusable Dishes and Utensils with Disposable Wet/Dry Wipes	Reusable Dishes and Utensils with Disposable Wet/Dry Wipes	Reusable Dishes and Disposable Utensils with Disposable Wet/Dry Wipes
PERSONAL HYGIENE	WASTE COLLECTION	FECAL COLLECTION	Dry John System	Apollo System	Skylab System
		URINE COLLECTION		Apollo System	Skylab System
		VOMITUS COLLECTION	Disposable Bags	Disposable Bags	Intimate Adaptor
	BODY CLEANSING	PARTIAL BODY WASHING	Disposable Wet Wipe	Disposable Wet Wipe	Skylab-Type Disposable Washcloth
		PARTIAL BODY DRYING	Disposable Dry	Disposable Dry	Electric Dryer
	PERSONAL GROOMING	SHAVING	Safety or Windup	Safety or Windup	Safety or Windup
		DENTAL CARE	Toothbrush w/Dentifrice	Toothbrush w/Dentifrice	Electric Toothbrush
HOUSEKEEPING	EQUIPMENT CLEANUP	SURFACE WIPING	Disposable Wet/ Dry Wipes	Disposable Wet/ Dry Wipes	Skylab-Type Disposable Cloth
	REFUSE MANAGEMENT	MANUAL COLLECTION	Disposable Trash Bag	Disposable Trash Bag	Disposable Recepticles
		VACUUM COLLECTION	Skylab-Type Electric	Vacuum-Vented	Skylab-Type Electric
		REFUSE DISPOSAL	Storage Bin/ Container	Storage Bin/ Container	Vacuum Storage
	GARMENT/LINEN MAINTENANCE	CLOTHES WASH/ DRY	Disposable Clothes	Disposable Clothes	Mechanical w/Clothes Line
OFF-DUTY ACTIVITIES	ENTERTAINMENT	MUSIC	Cassette Record Recorder	*	*
		LIBRARY	Books	*	*
		TELEVISION	Commercial Type	*	*
		GAMES	Cards, Handball, Etc.	*	*
	PHYSICAL CONDITIONING	EXERCISERS	Exer Gym, Hand Exerciser	*	*

* NOT TRADED

TABLE 2-2

MODULAR SPACE STATION APPLIANCE CONCEPTS TRADE STUDY RESULTS

HABITABILITY SUBSYSTEM	HABITABILITY FUNCTION	APPLIANCE FUNCTION	CONCEPT CHOSEN	FIRST RATED CONCEPT	SECOND RATED CONCEPT
FOOD MANAGEMENT	FOOD STORAGE	REFRIGERATED	Space Radiator	Space Radiator	Thermoelectric
		FROZEN	Space Radiator	Space Radiator	Thermoelectric
	FOOD PREPARATION	WARMING	Heating Trays	Heating Trays	Convective Oven
	GALLEY CLEANUP	DISH CLEANUP	Water Spray Wash/Elec. Heat Dry	Reusable Dishes and Disposable Wet/Dry Wipes	Reusable Cups & Dishes - Disposable Utensils and Disposable Wet/Dry Wipes
PERSONAL HYGIENE	WASTE COLLECTION	FECAL COLLECTION	Dry John System	Apollo System	Skylab System
		URINE COLLECTION	Dry John System	Apollo System	Skylab System
		VOMITUS COLLECTION	Disposable Bags	Disposable Bags	Intimate Adaptor
	BODY CLEANSING	SHOWER	Collapsible	Collapsible	Mechanical
		PARTIAL BODY WASHING	Reusable Wipes	Reusable Wipes	Skylab-Type Disposable Washcloths
		PARTIAL BODY DRYING	Reusable Wipes	Reusable Wipes	Disposable Dry Wipes
	PERSONAL GROOMING	SHAVING	Windup	Windup	Vacuum Driven
		HAIRCUTTING	Razor Comb Vacuum Collection	Razor Comb Vacuum Collection	Power Clipper Vacuum Collection
		NAIL CARE	Manual Clipper	Manual Clipper	Nail File Vacuum Collection
		DENTAL CARE	Toothbrush w/Dentifrice	Toothbrush w/Dentifrice	Electric Toothbrush
HOUSEKEEPING	EQUIPMENT CLEANUP	SURFACE WIPING	Reusable Wet/ Dry Wipes	Disposable Wet/ Dry Wipes	Sponge Skylab-Type
	REFUSE MANAGEMENT	MANUAL COLLECTION	Disposable Bags	Disposable Bags	Disposable Recepticles
		VACUUM COLLECTION	Skylab-Type (Electric)	Skylab-Type (Electric)	Vacuum Vented
		REFUSE PROCESSING	Compactor (Air Pressure)	Compactor (Air Pressure)	Compactor (Vacuum)
		REFUSE DISPOSAL	Storage Bin/ Container	Storage Bin/ Container	Vacuum Storage
	GARMENT/LINEN MAINTENANCE	CLOTHES WASH/DRY	Water Spray Agitation Plus Electric Dry	Disposable Clothes	Water Spray Agitation Plus Clothes Line
OFF-DUTY ACTIVITIES	ENTERTAINMENT	MUSIC	Casette Recorder	*	*
		LIBRARY	Books	*	*
		TELEVISION	Commercial Type	*	*
		GAMES	Cards, Handball; Etc.	*	*
	PHYSICAL CONDITIONING	EXERCISERS	Exer Gym, Hand Exerciser	*	*

3.0 SUBROUTINE OPERATION AND MATH MODEL DESCRIPTIONS

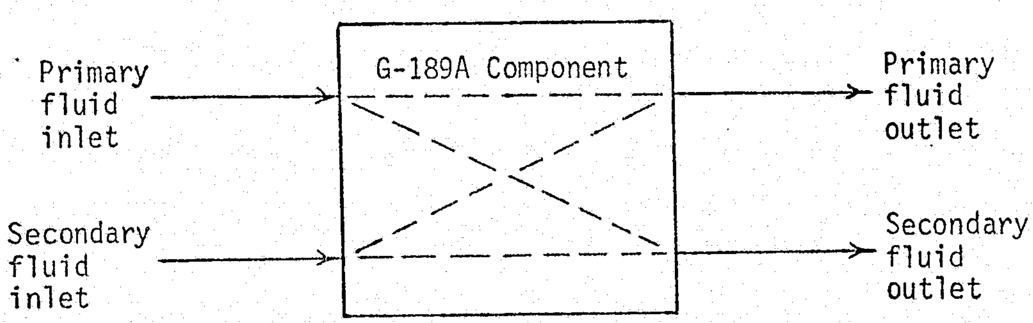
In this section are described (1) all user's input instructions required to operate the subroutines, (2) the analytical math model assumed to describe the component, and (3) the methods and logic used to obtain the solution. This section may serve independently as a user's manual for the subroutines. The symbols used in the subroutine descriptions are defined at the end of each section. The Fortran V listings of the subroutines are given in Appendix A. The logic flow chart for each subroutine, and the definition of the Fortran names used in the subroutine, are also given at the end of each section. The description of each subroutine is in the format prescribed in Reference 2 for new G-189A subroutine writeups as:

1.0 Subroutine Description
2.0 Subroutine Data
- 2.1 General Notes
- 2.2 Instruction Options
- 2.3 Heat Loss V-Array Data
- 2.4 Steady-State K-Array Data
- 2.5 Steady-State V-Array Data
- 2.6 Transient K-Array Data
- 2.7 Transient V-Array Data
- 2.8 Extra K-Array Data
- 2.9 Extra V-Array Data

3.0 Analytical Model Description

Subsections not required for a particular subroutine are omitted.

It is assumed that the user of these subroutines is familiar with the basic operation of the G-189A program. Every G-189A component has two possible fluid flow paths through it, as shown below. One side is arbitrarily



3.0 (Continued)

designated as primary and the other secondary. The description of each component must specify which flow paths are used. For example, for the SHOWER component, the primary inlet is used for the air flow, the secondary inlet is water, and the air/water mixture is outlet on the primary side. The secondary outlet for the SHOWER is not used.

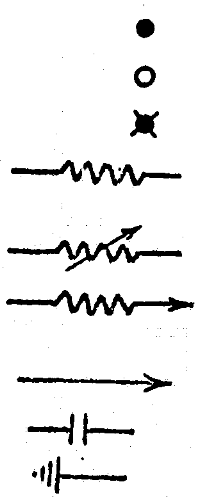
All the data for every G-189A component are stored in a V-array (which is equivalenced to a K-array to allow integer data). During solution of a particular component, its individual floating point data are loaded from the V-array into an R-array. The component subroutine then operates on these data in a manner which reflects the performance of that component. Thus, the data input descriptions in the following sections are concerned primarily with the definition of each K and R-array location for each component. Some of these data locations are used to store input data, and others output data. Thus the following convention has been followed to define the type of data for each component reference location:

Data Type	Meaning
I(R)	Input data - required
I(O)	Input data - optional
O	Output (computed) data
I(R), O	Initial input data required; thereafter data is computed and output

Default values for the model input data are built into each subroutine to describe specific appliances selected in the trade studies of Reference 1. For the refrigerator/freezer subroutine, the default values describe the Shuttle freezer kit design described in Reference 32. The user may therefore simulate these specific appliances with a minimum of input; or he may simulate other related appliances with the same subroutines by supplying new input data, which would then override the default values. Data reference locations for which default input data are included in the subroutine are designed by I(O) in the writeups. The default input values are listed in a table at the end of the model description for each subroutine.

3.0 (Continued)

A major part of each appliance subroutine is the thermal model used to simulate the heat transfer within the appliance and between the appliance and its surroundings. An equivalent electrical resistor/capacitor nodal network was used in each case for this purpose. These nodal models are shown for the various appliances in the following sections using these symbols:



Node with thermal mass

Steady-state node

Boundary node

Thermal conduction or convection conductor (linear)

Thermal radiation conductor (nonlinear)

One-directional fluid flow conductor

Other heat addition

Thermal capacitance

Thermal ground

Nodes having thermal mass are solved by equating the net heat input to the change in heat storage. A thermal capacitance is specified for these nodes, as defined by the relation

$$C = mc_p$$

where

C = nodal thermal capacitance

m = nodal mass

c_p = specific heat

Steady-state nodes are used to model gaseous fluid or other special nodes having negligible thermal mass. These nodes are in thermal equilibrium with their surroundings; that is, their temperature is computed such that the heat in is equal to the heat out. Boundary node temperatures are not computed in the subroutines. They must be input by the user and may be held constant or varied during a run based on the progressing solution.

3.0 (Continued)

The nodes are interconnected by thermal conductors evaluated in the following ways:

$$G = \begin{cases} kA/_{\ell} & \sim \quad \text{solid conduction} \\ h_c A_s & \sim \quad \text{fluid convection} \\ \dot{m} c_p & \sim \quad \text{fluid flow} \\ \sigma A_s \mathcal{F} & \sim \quad \text{radiation} \end{cases}$$

where
- G = thermal conductor
- k = material thermal conductivity
- A = "window" area between nodes
- A_s = surface area of node
- ℓ = length between nodal centers
- $\dot{m}$ = fluid mass flow rate
- h_c = convection heat transfer coefficient
- σ = Stefan-Boltzmann radiation constant
- $\mathcal{F}$ = radiation interchange factor

The first three conductors are referred to as linear and transfer heat proportional to the first power temperature difference $(T_j - T_i)$. The fourth conductor is radiation which transfers heat as a function of the fourth power temperature difference $(T_j^4 - T_i^4)$. Some conductors are designated as "one-directional" elements, meaning that heat is transferred through them in one direction only. This feature is typically used for fluid flow simulation, in which the stored energy travels downstream only, and also used for satisfying certain boundary conditions at a line of symmetry within a model.

The G-189A program is designed to run either with or without a system pressure drop analysis. If this option is not used, the system pressures will remain constant around each flow loop, except where they are set by the user when necessary by GPOLY logic. For problems requiring pressure

3.0 (Continued)

drop analysis, a set of six standard pressure drop equations is provided within the G-189A program to model any component. These equations are described in the G-189A program manual, Reference 2, and are shown in Table 3-1 taken from that reference. They are considered adequate to describe the pressure drop characteristics of the new appliance subroutines. The methods and recommended pressure drop data are included in the following sections for each appliance subroutine.

TABLE 3-1

G-189A PRESSURE DROP OPTIONS

ΔP Option Code	Pressure Drop Equations Solved	K Array Data Req'd (PDK* Loadsheet)	V Array Data Req'd (PDPR or PDSE Loadsheet)
0 (default)	$\Delta P = 0$	———	———
1	* $\Delta P = \frac{4f\ L_e}{D_h} \left(\frac{w}{A_c}\right)^2 \frac{1}{2\rho\ g_c}$ (psf) where: A_c = Cross sectional area (ft^2) D_h = Hydraulic diameter (ft) f = Fanning friction factor (dimensionless) g_c = Newton's law conversion factor (lb_m - ft/lb_f - hr^2) L_e = Equivalent length of pipe or duct for pressure drop calculations (ft) w = Flow (lb/hr) ρ = Flow stream density (lb/ft^3) The fanning friction factor, f, is calculated using one of the following equations which are selected on the basis of the fluid stream Reynolds number, Re = $\frac{w\ D_h}{\mu\ A_c}$.	———	D_h, L_e, A_c If A_c is not input the flow passage is assumed to be circular and A_c is calculated as follows: $A_c = \frac{\pi}{4} (D_h)^2$

TABLE 3-1 (Continued)

G-189A PRESSURE DROP OPTIONS

ΔP Option Code	Pressure Drop Equations Solved	K Array Data Req'd (PDK* Loadsheet)	V Array Data Req'd (PDPR or PDSE Loadsheet
	$f = \frac{16}{Re}$ (0<Re<2100) $= \frac{0.0791}{Re^{0.25}}$ (2100<Re<100000) $= 0.0014 + \frac{0.125}{Re^{0.32}}$ (100000<Re<∞) *NOTE: If the fluid is gaseous and a Mach number >0.1 is calculated then ΔP option codes of 1 or 2 will use compressible flow equations and solve the pressure drop using a method similar to that described by Shapiro in Reference 5-1.		
2	* $\Delta P = K \left(\frac{w}{A_c}\right)^2 \frac{1}{2\rho\, g_c}$ (psf) where: K = Loss coefficient (dimensionless) A_c, g_c, w, ρ = As defined under ΔP option code 1	——	K, A_c

TABLE 3-1 (Continued)

G-189A PRESSURE DROP OPTIONS

ΔP Option Code	Pressure Drop Equation Solved	K Array Data Req'd (PDK* Loadsheet)	V Array Data Req'd (PDPR or PDSE Loadsheet)
3	$\Delta P = C \frac{w^n}{\rho}$ (psf) where: C = Constant coefficient (dimensionless) n = Exponential coefficient for flow (dimensionless) w,ρ = As defined under ΔP option code 1	———	C, n
4	$\Delta P = C_1 \Delta P$ (psf) where: C_1 = Conversion factor = 5.204 (psf/in. H_2O) for gases = 144.0 (psf/psi) for liquids	———	ΔP Input ΔP as (in. H_2O) for gas flows or (psi) for liquid flows.
5	$\Delta P = \frac{\text{Interpolated value for } \rho\Delta P}{C_1 \rho}$ where: C_1,ρ = As defined under ΔP option codes 1 and 4	Input table number for ρΔP vs. w data For gases: Dependent variable: ρΔP (lb - in. H_2O/ft^3) 1st Independent variable: w (lb/hr)	———

TABLE 3-1 (concluded)

G-189A PRESSURE DROP OPTIONS

ΔP Option Code	Pressure Drop Equation Solved	K Array Data Req'd (PDK* Loadsheet)	V Array Data Req'd (PDPR or PDSE Loadsheet)
		For liquids: Dependent variable: $\rho\Delta P$ (lb-psi/ft^3) 1st Independent variable: w (lb/hr)	
6	ΔP = (Interpolated value for ΔP) C_1 where: C_1 = As defined under ΔP option code 4	Input table number for ΔP vs w data **For gases:** Dependent variable: ΔP (in. H_2O) 1st Independent variable: w (lb/hr) **For liquids:** Dependent variable: ΔP (psi) 1st Independent variable: w (lb/hr)	———

3.1 CHILLR

The CHILLR subroutine is designated as G-189A No. 71.

3.1.1 Subroutine Description

The CHILLR subroutine will simulate a refrigerator or freezer cooled either by an externally chilled coolant (e.g., from a radiator coolant circuit) or by a self-contained refrigeration unit. The model is fairly generalized and may be used to simulate the following configurations:

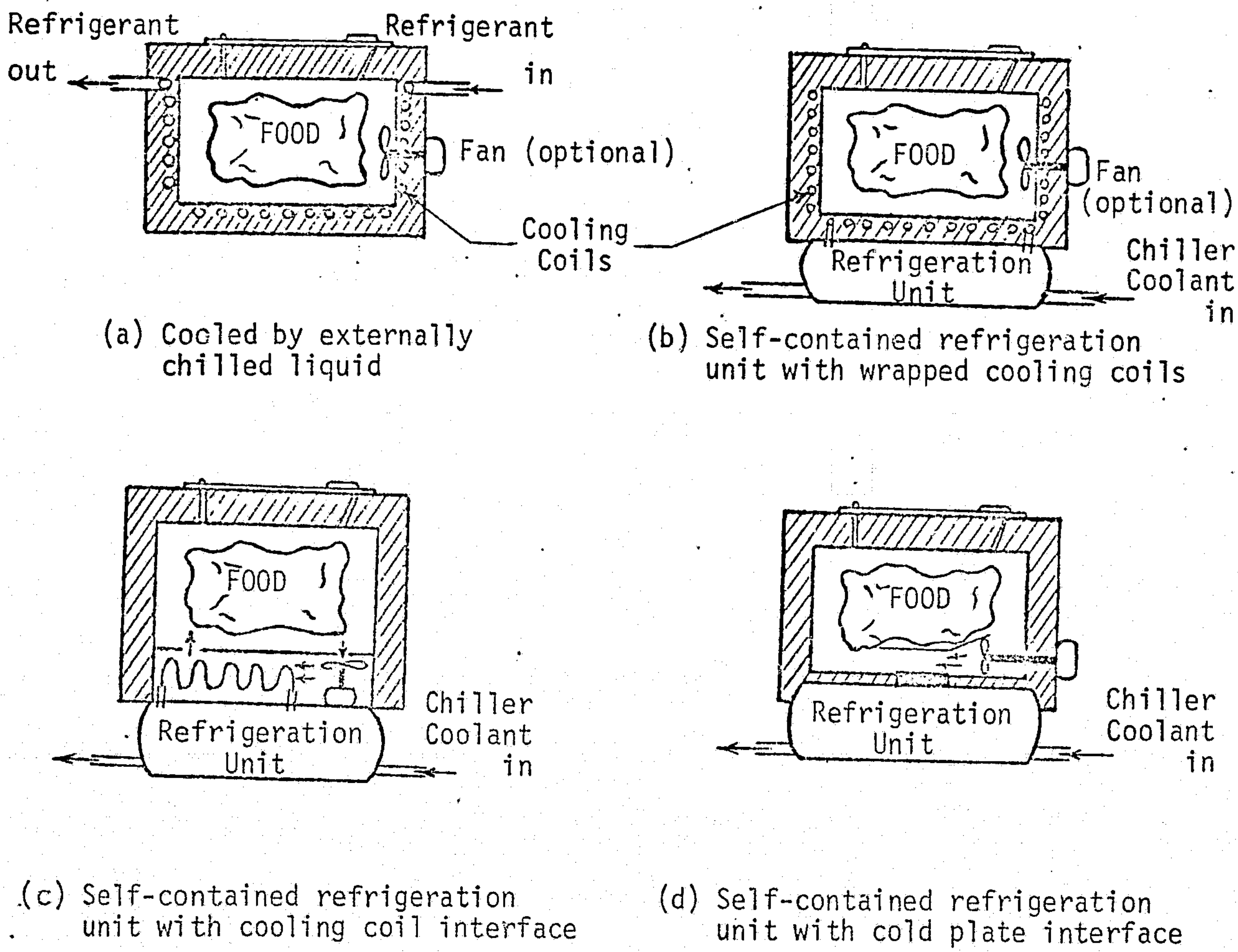


Figure 3-1. CHILLR Component and Flow Schematic

3.1.1 (Continued)

The thermal model in either case is shown in Figure 3-2. In addition to the thermal network shown in the figure, the locker inner walls and contents are thermally connected to the ambient surroundings using the standard G-189A subroutine QSURR. This routine models the heat exchange from an arbitrary structure to ambient via insulation, thermal shorts, and conduction/convection/radiation paths. The output from the QSURR subroutine defines the heat loss from the internal structure to ambient, which is designated as q_{surr} in Figure 3-2.

The CHILLR subroutine has been used to simulate the Shuttle food and medical sample freezer kit described in Reference 32. Excellent correlation between the subroutine results and independent detailed freezer thermal analysis has been obtained, as presented in Section 4.1. The model input data used for that freezer design are included directly in the subroutine as default data, and are listed in Table 3-2.

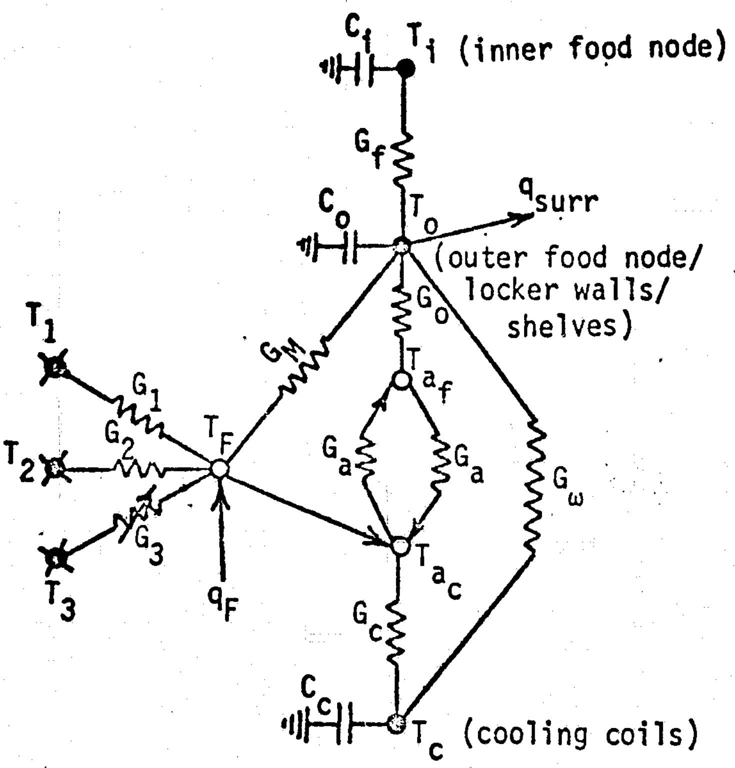
3.1.2 Subroutine Data

3.1.2.1 General Notes

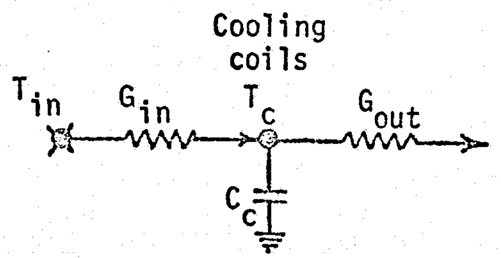
a. Only the component primary side is used. If an externally chilled coolant is assumed, the coolant must inlet and exit on the primary side. If a self-contained cooling unit is used, the primary fluid is assumed to cool the heat rejection device at the hot side of the unit.

b. Any flow codes may be used, but no fluid phase change is allowed.

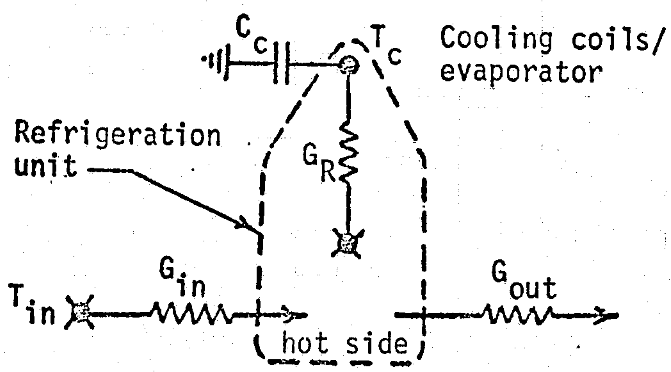
c. If ambient air properties (for evaluating door opening effects) are taken from an associated cabin component [an option with instruction NSTR(4)], then the cabin component number must be included on the KBAS card for the refrigerator or freezer. The CHILLR routine uses cabin V-array locations 1 through 19.



(a) Thermal model of refrigerator/freezer locker including cooling coils



(b) Thermal model of refrigerator/freezer chiller side, including cooling coils - assuming externally chilled coolant fluid



(c) Thermal model of refrigerator/freezer chiller side, including cooling coils - assuming self-contained cooling unit

Figure 3-2. Thermal Model of Refrigerator/Freezer Locker (a) and Chiller (b) and (c)

3.1.2.1 (Continued)

d. The net heat loss from the refrigerator or freezer locker, locker fan, and door opening effects to the ambient surroundings should be added to the associated cabin sensible heat input. This heat loss is given in R(53), and should be added to cabin R-array location 66 in GPOLY logic.

3.1.2.2 Instruction Options

NSTR(1): Flag used to designate type of refrigerator or freezer being used

= 0 No self-contained refrigeration unit is used. Unit is cooled by chilled liquid inlet to primary side from external coolant circuit.

= 1 A self-contained refrigeration unit is used, with the heat rejection device cooled by the primary side flow.

NSTR(2): Control logic for fan inside refrigerator or freezer locker

= 0 No fan is used.

= 1 Fan is on continuously if transient case, or for fraction of time in R(93) if steady state.

= 2 Fan is controlled ON/OFF with the self-contained cooling unit according to NSTR(3) [transient case with NSTR(1) = 1 only].

NSTR(3): Temperature control logic (used in transient case only). If a self-contained cooling unit is used, this will control the cooling unit ON/OFF cycle; and also the locker fan ON/OFF cycle if NSTR(2) = 2. If an external coolant circuit is used, this will control the fan ON/OFF cycle if NSTR(2) = 2.

3.1.2.2 (Continued)

= 0 Previous components turned ON/OFF based on user's input control parameter in R(113).

= 1 Previous components turned ON/OFF based on locker internal temperature limits between R(83) (low temperature to turn components OFF) and R(84) (high temperature for components to turn ON).

= 2 Previous components cycled ON/OFF according to a timer cycle in R(110) through R(112).

NSTR(4): Identifies ambient gas properties for door opening effects

= 0 Use cabin component NCAB (in K-array location 8)

= 1 Use gas properties input in R(96) through R(99)

NSTR(5): Door opening schedule

= 0 Input average number of times door is opened per day [R(71)] and mission time for first opening in R(114)

= 1 Input arbitrary daily door opening schedule in extra V-array location beginning with R(125)

NSTR(6): Flag used to identify type of refrigeration unit (used only if NSTR(1) = 1)

= 0 Stirling cycle

= 1 Vapor compression

3.1.2.3 Heat Loss V-Array Data

Reference Location	Description	Data Type
51	Temperature of locker internal walls and food outer surface (°F)	I(0), 0
52	Summed thermal conductance from locker internal walls to ambient surroundings, excluding conductance through locker fan (Btu/hr °F)	0
53	Net heat loss from locker internal surface, from fan (if present), and from door opening to ambient surroundings (Btu/hr)	0
54	Ambient gas temperature (°F)	I(0)
55	Thermal conductor from refrigerator or freezer locker external surface (not including fan) to ambient gas (Btu/hr °F)	I(0)
56	Convective heat loss from locker outer surface (excluding fan) to ambient gas (Btu/hr)	0
57	Ambient wall temperature (°F)	I(0)
58	Thermal radiation conductor (A∃) between locker external surface and ambient walls (sq ft)	I(0)
59	Radiative heat loss from refrigerator or freezer locker to ambient walls, excluding fan (Btu/hr)	0
60	Temperature of ambient structure attached to locker (°F)	I(0)
61	Thermal conductor through structural hard-attach-points between locker inner walls and attached ambient structure, excluding conductance through insulation (Btu/hr °F)	I(0)
62	Thermal loss through structural hard-attach-points (Btu/hr)	0

3.1.2.3 (Continued)

Reference Location	Description	Data Type
63	Refrigerator or freezer locker external surface temperature (°F)	0
64	Thermal conductor through locker wall insulation (Btu/hr °F)	I(0)

3.1.2.4 Steady-State and Transient V-Array Data

Note: Locations used for transient runs only are marked with an asterisk.

Reference Location	Description	Data Type
65	Net cooling provided to refrigerator or freezer locker (Btu/hr)	0
66	Set point temperature; used to determine steady-state duty cycle for Stirling cycle cooling unit (°F)	I(0) if steady state, NSTR(1) = 1 and NSTR(6) = 0
67	Steady state duty cycle used for self-contained cooling unit (fraction of time on)	I(0) if NSTR(1) = 1 0 if NSTR(6) = 0
68	Total internal locker volume including food (cu ft)	I(0)
69	Packaged food volume (cu ft)	I(0)
70	Air change per door opening (fraction of void volume)	I(0)
71	Number of times per day door is to be opened	I(0) if steady state or NSTR(5) = 0
*72	Dry food mass (lbs)	I(0)
*73	Fraction of total food mass assigned to food outer surface node	I(0)
*74	Refrigerator or freezer locker inner shell thermal mass (Btu/°F)	I(0)
75	Food inner node temperature (°F)	I(0), 0

3.1.2.4 (Continued)

Reference Location	Description	Data Type
76	Ratio of frozen food thermal conductivity to unfrozen food thermal conductivity	I(0) for freezer only
77	Air temperature inside food compartment (°F)	I(R) if NSTR(2)≠0, 0
78	Thermal conductor, food inner node to outer node, in unfrozen condition (Btu/hr °F)	I(0)
79	Air temperature inside chilling compartment (°F)	I(R) if NSTR(2)≠0, 0
80	Surface area of cooling coils (sq ft)	I(R) if NSTR(2)≠0
81	Surface area of food and inner locker walls (sq ft)	I(R) if NSTR(2)≠0
82	Conductive thermal conductor from cooling coils to surface of food and inner locker walls (Btu/hr °F)	I(0)
*83	Food surface lower temperature limit for cooling unit and/or fan to turn off(°F)	I(0) if NSTR(3)=1
*84	Food surface high temperature limit for cooling unit and/or fan to turn on (°F)	I(0) if NSTR(3)=1
85	Convective heat transfer coefficient; air inside locker to food and walls inner surface and cooling coils (Btu/hr-sq ft-°F)	I(R) if NSTR(2)≠0
86	Conductive thermal conductor from fan to attached ambient structure (Btu/hr °F)	I(R) if NSTR(2)≠0
87	Effective fan housing temperature (°F)	I(R) if NSTR(2)≠0, 0
88	Thermal radiation conductor AF from fan housing to ambient walls (sq ft)	I(R) if NSTR(2)≠0

3.1.2.4 (Continued)

Reference Location	Description	Data Type
89	Conductive thermal conductor from fan housing to inner locker walls and food surface (Btu/hr °F)	I(R) if NSTR(2)≠0
90	Convective thermal conductor from fan housing to ambient gas (Btu/hr °F)	I(R) if NSTR(2)≠0
91	Fan motor input electrical power (watts)	I(R) if NSTR(2)≠0
92	Fan motor efficiency (fraction)	I(R) if NSTR(2)≠0
93	Fraction of time fan is on during steady-state run	I(R) for steady state
*94	Combined thermal mass of food outer surface node and inner locker walls (Btu/°F)	0
*95	Food inner node thermal mass (Btu/°F)	0
96	Inlet gas temperature when door is opened (°F)	I(0) if NSTR(4)=1
97	Inlet gas density when door is opened (lb/cu ft)	I(0) if NSTR(4)=1
98	Inlet gas absolute humidity when door is opened (lb water vapor/lb dry air)	I(0) if NSTR(4)=1
99	Inlet gas specific heat when door is opened (Btu/lb °F)	I(0) if NSTR(4)=1
100	Coefficient of performance of refrigeration unit (Btu cooling provided/BTU input electrical power); if NSTR(6)=0, this should not include coolant pump power	I(0) if NSTR(1)=1
101	Air circulation flow rate provided by locker fan (cfm)	I(R) if NSTR(2)≠0
102	Constant evaporator temperature for self-contained refrigeration device (°F)	I(R) if NSTR(1)=1 and NSTR(6)=1
*103	Cooling coils thermal mass-dry (Btu/°F)	I(0)
104	Cooling coils temperature (°F)	I(0), 0

3.1.2.4 (Continued)

Reference Location	Description	Data Type
105	Heat added into locker by opening door on previous iteration (Btu/hr)	0
106	Total cooling capacity of Stirling cycle cooling unit (Btu/hr)	I(0) if NSTR(1)=1, NSTR(6)=0
107	Coolant pump power used with Stirling cycle cooling unit (watts)	I(0) if NSTR(1)=1, NSTR(6)=0
108	Fraction of coolant pump power transferred to coolant with Stirling cycle unit	I(0) if NSTR(1)=1, NSTR(6)=0
109	Total electrical power used on previous iteration by self-contained cooling unit (watts)	0
*110	Preset time for cooling unit and/or fan to remain on with timer (minutes)	I(R) if NSTR(3)=2
*111	Preset time for cooling unit and/or fan to remain off with timer (minutes)	I(R) if NSTR(3)=2
*112	Initial displacement of time into timer ON/OFF cycle (minutes) Note: The cycle begins with the ON setting.	I(R) if NSTR(3)=2
*113	User's control parameter to turn on cooling unit and/or fan. Set to 1.0 when these components are to be turned on, or 0.0 when they are to be off.	I(R) if NSTR(3)=0
*114	Next mission time which locker door shall be opened (seconds)	I(R) if NSTR(5)=0, 0
*115	Total water content (frozen and unfrozen) in outer food node (lb water/lb total food mass in outer food node)	I(0)
*116	Total water content (frozen and unfrozen) in inner food node (lb water/lb total food mass in inner food node)	I(0)

3.1.2.4 (Continued)

Reference Location	Description	Data Type
*117	Fraction of water in outer food node that is frozen	I(R) only if food temperature initialized at 32.0°F, 0
*118	Fraction of water in inner food node that is frozen	I(R) only if food temperature initialized at 32.0°F, 0
*119	Specific heat of dry food (Btu/lb °F)	I(0)
120	Conductive thermal conductor from cooling coils to self-contained cooling unit structure (Btu/hr °F)	I(0)
121	Cooling unit structure temperature attached to conductor R(120) (°F)	I(0)
122-124	Not used	

3.1.2.5 Transient K-Array Data

Reference Location	Description	Data Type
16	R-array location to be used for next mission time locker door will be opened; equal to 125 or greater.	I(0) if NSTR(5)=1, 0
17	This location stores cooling unit mode on previous iteration: 0=OFF, 1=ON. Note: This location used to specify initial ON/OFF mode for cooling unit.	I(R) if NSTR(3)=1, 0
18	Not used	
19	Number of mission days for which solution has been obtained; used in door opening schedule logic	0

3.1.2.6 Extra V-Array Data (If NSTR(5)=1, there must be reserved N+1 locations where N is the number of locker door openings per day.)

Reference Location	Description	Data Type
125—>124+N	In these locations are stored the mission times (hrs) when the locker door is to be opened, where N is the number of door openings per day. Each opening is automatically repeated every 24 hrs.	I(R) if NSTR(5)=1, 0
125+N	Must be set to zero to indicate the end of door opening time entries	I(R) if NSTR(5)=1

3.1.3 Analytical Model Description

3.1.3.1 Fan Thermal Balance

Use of a fan to circulate the locker air is optional according to the input instruction NSTR(2). If the fan is omitted, all the thermal conductors connected to it are simply ignored in the model. If a fan is

3.1.3.1 (Continued)

included, its electrical input power (q_F) and motor efficiency (η) must be input. The heat generated within the fan motor is then given by

$$q_M = (1 - \eta)\, q_F \qquad (3.1.1)$$

(See Table 3-1 for definition of symbols.) The output shaft power is assumed to be dissipated as heat of compression of the inside locker air according to the relation:

$$q_a = \eta q_F \qquad (3.1.2)$$

The fan motor/housing is assumed to be a steady-state node; that is, its thermal mass is neglected and it is in thermal equilibrium with its surroundings (heat in = heat out). An energy balance on the fan is therefore as follows:

$$q_F = q_a + G_M(T_F-T_o) + G_1\,(T_F-T_1) + G_2\,(T_F-T_2) + \sigma G_3\,(T_F{}^4-T_3{}^4) \qquad (3.1.3)$$

where the temperatures in the radiation term are in °R. To solve equation (3.1.3) for fan temperature T_F, the radiation term is first linearized as follows:

$$\sigma G_3\,(T_F{}^4-T_3{}^4) = \underbrace{\sigma G_3\,(T_F{}^2+T_3{}^2)\,(T_F+T_3)}_{\equiv Y}\,(T_F-T_3) \qquad (3.1.4)$$

The effective radiation conductor Y is assumed to remain constant over a single time step (or steady-state iteration). Thus, it may be evaluated first, and the fan temperature obtained by solving equations (3.1.3) and (3.1.4):

$$T_F = \frac{q_F - q_a + G_M T_o + G_1 T_1 + G_2 T_2 + Y T_3}{G_M + G_1 + G_2 + Y} \qquad (3.1.5)$$

3.1.3.1 (Continued)

If the fan is turned off, or if there is no fan present, the convection conductors G_a, G_c, and G_o in Figure 3-2 are assumed zero. The air temperatures in the food and chilling compartments are set to the food temperature and cooling coils temperature, respectively. When the fan is operating, these temperatures are computed by equating the heat into and out of the air nodes, as follows:

$$G_o \left(T_o - T_{a_f}\right) = G_a \left(T_{a_f} - T_{a_c}\right) \tag{3.1.6}$$

$$q_a + G_a \left(T_{a_f} - T_{a_c}\right) = G_c \left(T_{a_c} - T_c\right) \tag{3.1.7}$$

Solving equations (3.1.6) and (3.1.7) for the locker air temperatures, one obtains:

$$T_{a_c} = \frac{(q_a + G_c T_c)\ (G_a + G_o) + G_a G_o T_o}{G_c\ (G_a + G_o) + G_a G_o} \tag{3.1.8}$$

$$T_{a_f} = \frac{G_o T_o + G_a T_{a_c}}{G_o + G_a} \tag{3.1.9}$$

3.1.3.2 Storage Compartment Thermal Solution

The steady-state nodes in the model (assumed to have negligible thermal mass) are all connected with the fan, as seen in Figure 3-2, and their solution was discussed in Paragraph 3.1.3.1. The remaining nodes normally have thermal mass, although it is neglected when obtaining a steady-state solution. Their solution is obtained by a standard forward difference method as follows:

$$T_{new} = T_{old} + \gamma\ \Sigma\ q_{into\ node} \tag{3.1.10}$$

(One exception to this equation is the cooling coils temperature T_c when a vapor compression unit is operating, as discussed in Section 3.1.3.4). For

3.1.3.2 (Continued)

a transient solution, the value of γ is given by

$$\gamma_{transient} = \frac{\Delta t}{mc_p} \qquad (3.1.11)$$

The value of mc_p is simply the thermal mass of the node. The computing time increment Δt is taken to be 0.4 times the smallest of the time increments required for computational stability, where

$$\Delta t_{stability} = \frac{\text{thermal mass}}{\text{sum of conductors to ground}} \qquad (3.1.12)$$

If the stability time increment is larger than the G-189A system model time step, then the system time step is used in equation (3.1.11). If the stability time increment is smaller than the system time step, then internal iterations are performed until the required system time step is achieved.

For the steady-state solution, the value used for γ in equation (3.1.10) is given by

$$\gamma_{steady\ state} = \frac{0.4}{\text{sum of conductors to ground}} \qquad (3.1.13)$$

For steady-state solutions, internal iterations of these nodes using equations (3.1.10) and (3.1.13) are continued until the old and new temperatures differ by less than 0.001°F, or until 25 iterations are made.

3.1.3.3 Freezing Effects

The thermal properties of the food nodes are assumed to vary with the water/ice balance. When food temperatures drop to 32°F, their temperature is held constant until the latent heat of their moisture is removed. Thawing of frozen food was considered to represent a failure condition and was not included in the model. The thermal conductivity of unfrozen food

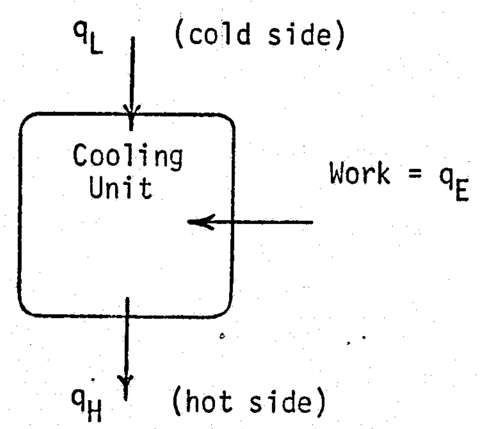
3.1.3.3 (Continued)

is assumed until all the liquid water has frozen, after which the conductivity of frozen food is used. The thermal mass of the food nodes (in the transient case) is computed as follows:

$$(mc_p)_{\text{food}} = (mc_p)_{\substack{\text{bone-dry}\\ \text{food}}} + m_{\substack{\text{liquid}\\ \text{water}}} + 0.46\, m_{\text{ice}} \qquad (3.1.14)$$

3.1.3.4 Self-Contained Refrigeration Unit

If the input instruction NSTR(1) is set to 1, a self-contained refrigeration unit is assumed as follows:



Logic for two types of refrigeration units is included in the subroutine, depending on the input value of NSTR(6). If NSTR(6)=0, a Stirling cycle is assumed. This type of unit has been selected for the conceptual design of a Shuttle food and medical sample freezer, described in Reference 32. If NSTR(6)=1, a vapor compression unit is assumed, with a compression/condensation/expansion/evaporation cycle. For both cases, the coefficient of performance of the unit, defined as follows, must be input:

$$\text{COP} \equiv \frac{q_L}{q_E} \qquad (3.1.15)$$

3.1.3.4 (Continued)

where q_L is the total cooling provided to the locker, and q_E is the electrical power input to the unit. For the vapor compression unit, the coolant pump, if present, is assumed to cycle on and off with the refrigeration unit, and the electrical power q_E should include the pump power. To be compatible with the Shuttle freezer design, the coolant pump used with the Stirling cycle is assumed to be on continuously. It is therefore treated separately from the Stirling unit, and the pump electrical power should not be included in q_E. The heat rejected at the freezer hot side is given by

$$q_H = q_L + q_E + (1 - j)q_P \tag{3.1.16}$$

where q_P is the coolant pump power for the Stirling cycle (or zero for vapor compression), and j is the fraction of the pump power transferred to the coolant. Combining equations (3.1.15) and (3.1.16), the total heat rejected by the cooling unit to the environment is given by

$$q_H = q_E\,(1 + COP) + (1 - j)q_P = q_L(1 + \frac{1}{COP}) + (1 - j)q_P \tag{3.1.17}$$

This rejected heat is all put into the primary flow which is assumed to be directed to the chiller unit heat rejection device. Thus, the primary flow outlet temperature is given by

$$T_{primary\ out} = T_{primary\ in} + \left[\frac{q_E\,(1 + COP) + (1 - j)q_P}{(\dot{m}c_p)_{primary\ flow}}\right] \tag{3.1.18}$$

For the vapor compression cycle, the cooling coils temperature T_c is assumed to remain constant when the unit is turned on, based on the design operating pressure of the working fluid. For the Stirling cycle, the rate of heat removed from the cooling coils is assumed to be constant when the unit is on.

For steady state runs using a self-contained refrigeration unit, an equipment duty cycle is input to define the fraction of time the unit is

3.1.3.4 (Continued)

turned on. The total cooling provided, q_L , and electrical power, q_E , are multiplied by this fraction before using in equations (3.1.16) through 3.1.18). For the Stirling cycle case, the duty cycle is internally adjusted to give the required control temperature input in R(66). These adjustments are made using the following relation:

$$D_{new} = D_{old}\left[\frac{T_{ambient} - T_{control}}{T_{ambient} - T_{food}}\right]$$

$$= R(67)\left[\frac{R(54) - R(66)}{R(54) - R(51)}\right]$$

3.1.3.5 Locker Door Opening Effects

The heat input from air exchange through the locker door will depend on the temperature, density, specific heat and humidity of the ambient air. These properties may be input directly, or a cabin component may be used from which to obtain the data, as specified in input instruction NSTR(4). The door may be opened according to an arbitrary input schedule (transient runs only) or at constant periodic intervals, according to input instruction NSTR(5). The heat input from the door is given by

$$q_{door} = \rho_a V_{locker} f[c_{p_a}(T_{ambient} - T_o) + \alpha Q_{LAT}]/\Delta t \qquad (3.1.19)$$

where f is the fraction of locker void volume assumed to be exchanged with ambient air. The time Δt is the system computing time step for transient runs, and the average time between door openings for steady-state runs.

3.1.3.6 Pressure Drop Considerations

If a self-contained cooling unit is used, the only fluid flow connections within the ECLSS model will be to provide a fluid to cool the heat rejection device. Typically, this will involve either cabin air flow or a system

3.1.3.6 (Continued)

coolant flowing through some type of heat exchanger. In the former case, no pressure drop model data would be required for the CHILLR component since it would not be connected within an actual fluid loop. For the latter case, the specific pressure drop data to be used would depend on the geometry of the heat exchanger device. This has not yet been designed for a spacecraft refrigerator or freezer, and therefore the pressure drop model data cannot now be specified. Similarly, if an externally chilled fluid were used to cool the refrigerator or freezer, the geometry of the cooling coils inside the locker would have to be known in order to specify pressure drop model data. However, it is believed in both cases that probably the G-189A pressure drop option (1) in Table 3-1 would best describe the device. Only the flow path cross-sectional area, hydraulic diameter, and effective length would be input. The program would then determine the flow regime (laminar, transition, or turbulent) and compute the pressure drop accordingly.

TABLE 3-2

DEFAULT VALUES FOR CHILLR SUBROUTINE INPUT DATA

R-Array* Location	Default Value	R-Array Location	Default Value	
51	- 9.5	84	- 7	
54	70	96	70	
55	33.5	97	0.075	
57	70	98	0.0078	
58	11	99	0.24	
60	70	100	0.364	
61	0.2	103	3.8	
64	1.48	104	A(2)	NSTR(1)=0
66	- 8.5		-10	NSTR(1)=1, NSTR(6)=0
67	0.63		R(102)	NSTR(1)=1, NSTR(6)=1
68	4.63	106	75 x 3.412	
69	4.13	107	5	
70	1	108	1	
71	18	115	0.5	
72	135	116	0.5	
73	0.17	117	0	R(51) > 32
74	3.8		1	R(51) < 32
75	- 8.6	118	0	R(75) > 32
76	3.5		1	R(75) < 32
78	18.57	119	0.49	
82	260	120	0.129	
83	- 10.5	121	97	

K(16)=125

*See Section 3.1.2 for definition of variables.

TABLE 3-3

DEFINITION OF SYMBOLS FOR CHILLR SUBROUTINE DESCRIPTION

C_c = R(103) = Thermal mass of cooling coils-dry (Btu/°F)

C_i = R(95) = Thermal mass of food inner node (Btu/°F)

C_o = R(94) = Combined thermal mass of food outer surface node and inner locker walls/shelves (Btu/°F)

COP = R(100) = Coefficient of performance of refrigeration unit (Btu cooling provided/Btu input electrical power)

C_p = Specific heat (Btu/lb °F)

C_{p_a} = Specific heat of air (Btu/lb °F)

D = R(67) = Operational duty cycle for self-contained refrigeration unit (fraction of time on)

f = R(70) = Fraction of locker void volume which is exchanged with ambient air when door is opened

G_a = $R(101)\,\rho_a \times 60\left(\frac{min}{hr}\right) C_{p_a}$ = Thermal conductor ($\dot{m}\,C_p$) of air between chilling and storage compartments (Btu/hr °F)

G_c = R(85) x R(80) = Thermal convection conductor between air in chilling compartment and cooling coils (Btu/hr °F)

G_f = R(78) - unfrozen food; R(78) x R(76) - frozen food = Thermal conductor between inner and outer food nodes (Btu/hr °F)

G_M = R(89) = Thermal conductor from fan motor to food outer surface and locker shelves/walls (Btu/hr °F)

G_o = R(85) x R(81) = Thermal convection conductor between air in food compartment and food/walls outer surface (Btu/hr °F)

G_w = R(82) = Thermal conductor from cooling coils to food outer surface and locker shelves/walls (Btu/hr °F)

G_1, G_2, G_3 = R(89), R(90), R(88) = Thermal conductor between fan and ambient surroundings by conduction (Btu/hr °F), convection (Btu/hr °F) and radiation (sq ft), respectively

j = R(108) = Fraction of coolant pump power transferred to coolant with Stirling cycle unit

m = mass (lbs)

TABLE 3-3 (continued)

DEFINITION OF SYMBOLS FOR CHILLR SUBROUTINE DESCRIPTION

$\dot{m}$ = Mass flow rate (lb/hr)

q_a = Heat input to locker air from fan (Btu/hr)

q_E = Electrical power input to the self-contained refrigeration unit (Btu/hr)

q_F = Total electrical power input to fan motor (Btu/hr)

q_H = Total heat rejection rate to the environment by the self-contained refrigeration unit (Btu/hr)

q_L = Cooling rate provided to the locker by the self-contained refrigeration unit (Btu/hr)

q_M = Heat generated within the fan motor (Btu/hr)

q_P = R(107) x 3.413 = Coolant pump input electrical power used with Stirling cycle unit (Btu/hr)

q_{surr} = Heat loss from locker internal surface and food to ambient surroundings through the locker walls, excluding heat transfer to the fan (Btu/hr)

Q_{LAT} = Latent heat of moisture (condensation and/or fusion) in air entering locker door (Btu/lb)

T_{a_c} = R(79) = Air temperature inside chilling compartment (°F)

T_{a_f} = R(77) = Air temperature inside food compartment (°F)

T_c = R(104) = Cooling coils temperature (°F)

T_F = R(87) = Effective fan housing temperature (°F)

T_i = R(75) = Temperature of food inner node (°F)

T_o = R(51) = Temperature of food outer node and locker internal shelves/walls (°F)

T_{in} = A(2) = Inlet fluid temperature to cooling coils (for an externally chilled unit) or to chiller heat rejection device (for a self-contained refrigeration unit); (°F)

TABLE 3-3 (concluded)

DEFINITION OF SYMBOLS FOR CHILLR SUBROUTINE DESCRIPTION

T_1, T_2, T_3 = Temperatures of ambient surroundings connected to fan by conduction, convection, and radiation, respectively (°F unless otherwise specified)

Y = Effective radiation conductor between fan and ambient wall, linearized according to equation (3.1.4)

Δt = Computing time step (hours)

α = Humidity from ambient air condensed inside locker when door is opened (lb water vapor/lb dry air)

γ = Multiplier to compute new temperatures, defined by equation (3.1.10)

η = Fan motor efficiency (fraction)

σ = Stefan-Boltzmann constant (0.1714×10^{-8} Btu/hr-sq ft-°R^4)

ρ_a = Density of air (lb/cu ft)

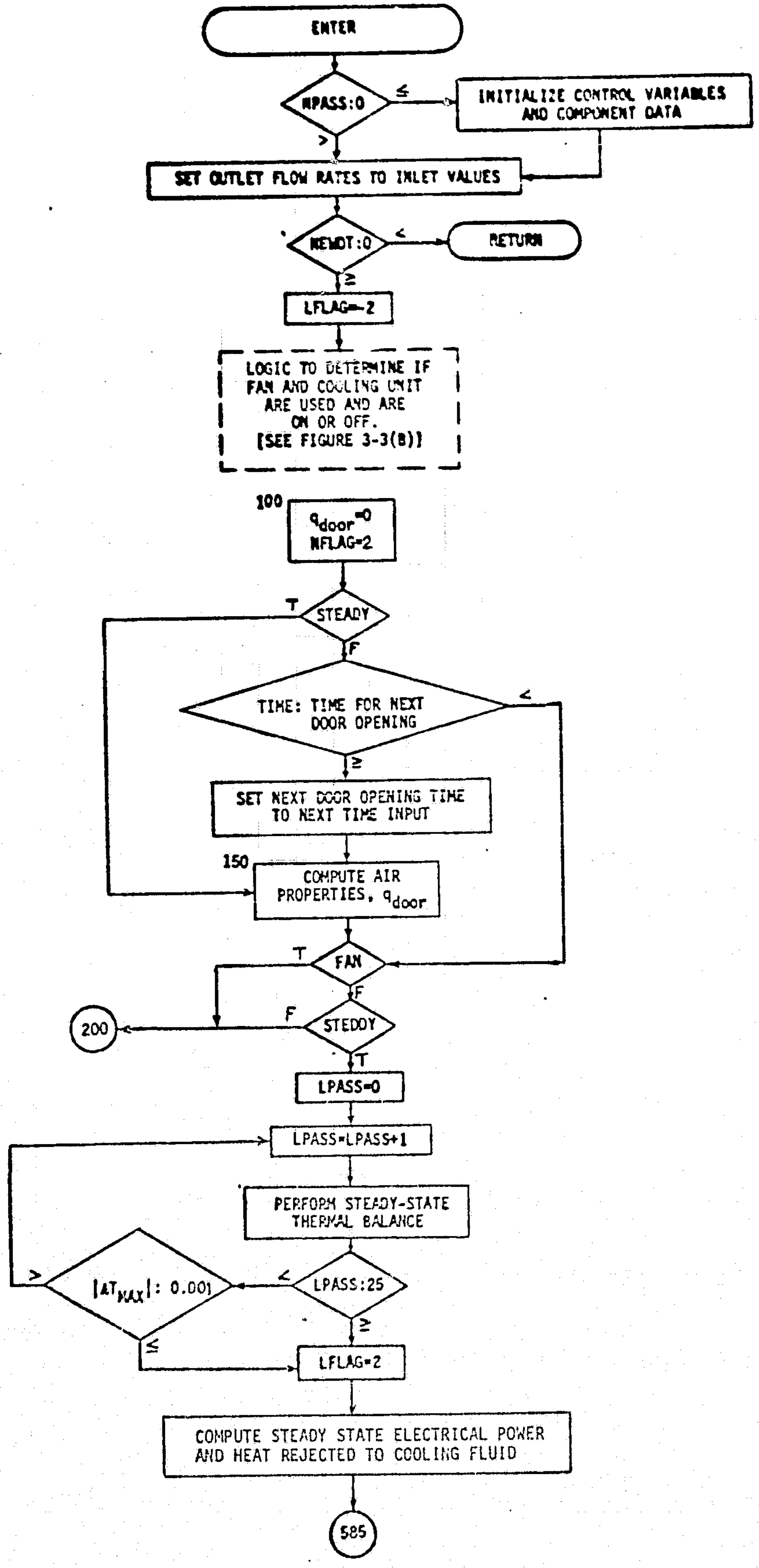


Figure 3-3. Logic Flow Chart for CHILLR Subroutine
(a) Main Subroutine Logic

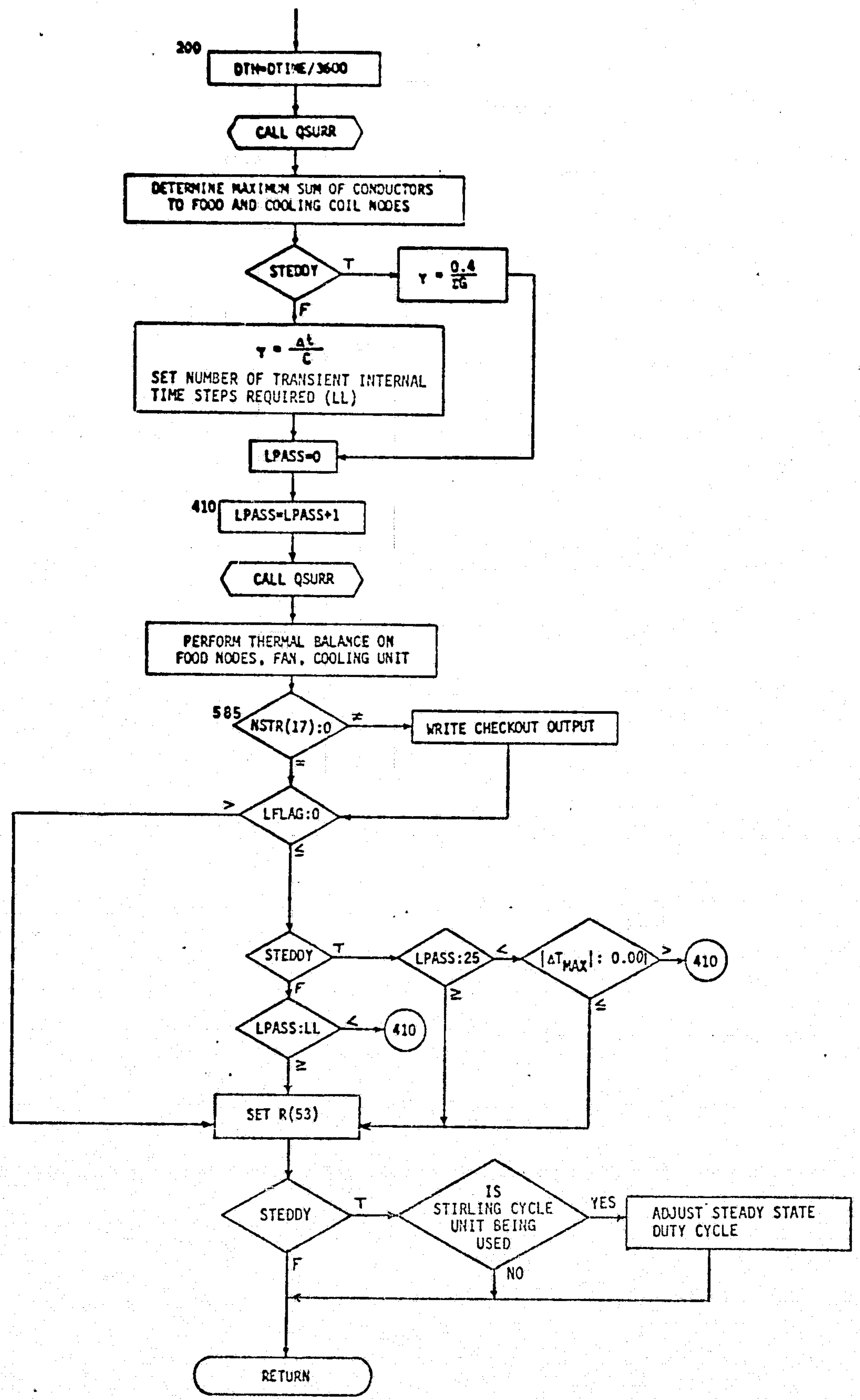


Figure 3-3. Logic Flow Chart for CHILLR Subroutine (continued)
(a) Main Subroutine Logic

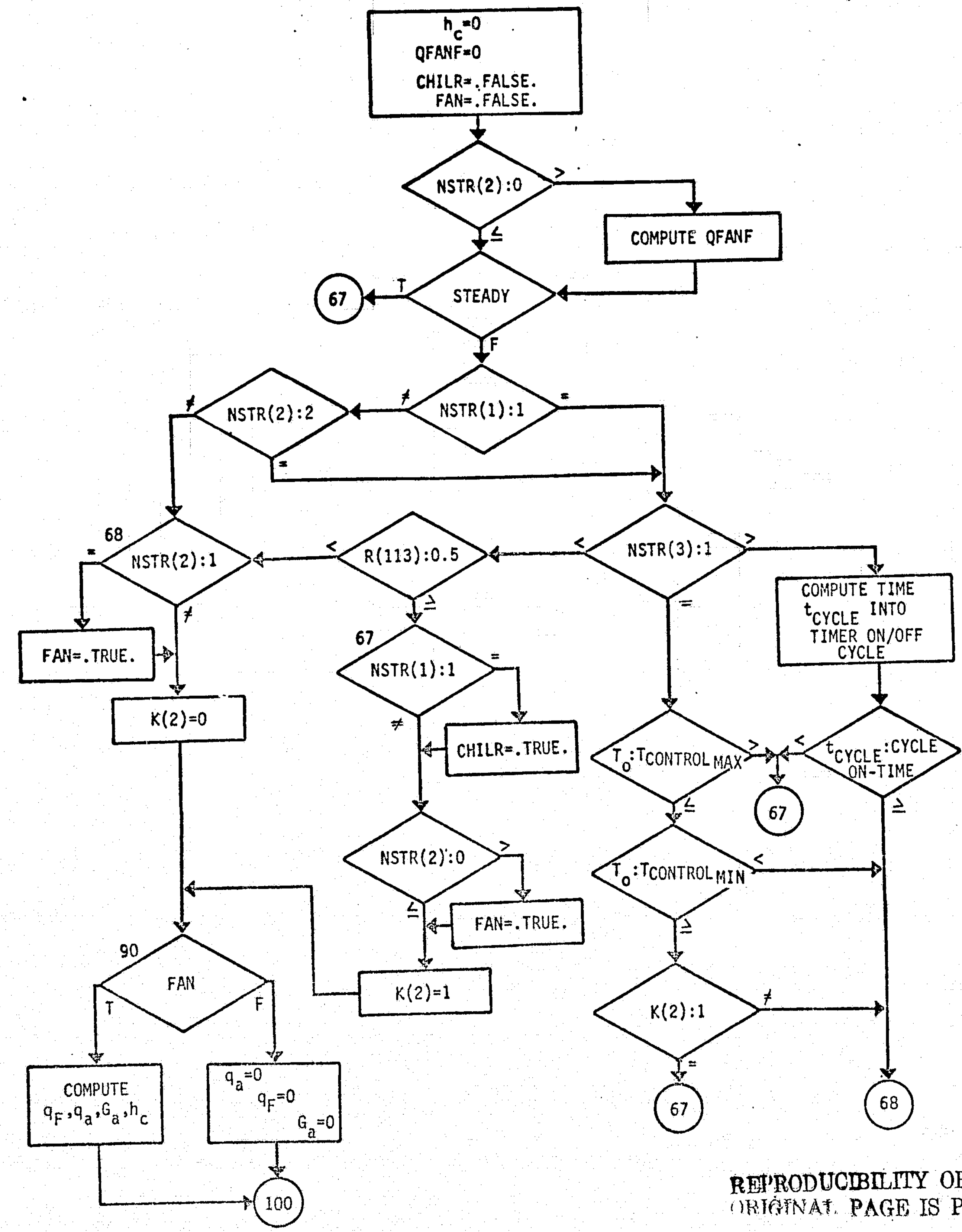


Figure 3-3. Logic Flow Chart for CHILLR Subroutine (concluded)

(b) Fan and Cooling Unit Control Logic

TABLE 3-4

DEFINITION OF CHILLR SUBROUTINE VARIABLES

C2	=	Thermal mass of cooling coils (or other cold sink); (Btu/°F)
CAIR	=	Thermal mass flow rate ($\dot{m}\ C_p$) of air circulated by food locker fan (Btu/hr °F)
CFOODI	=	Thermal mass of inner food node (Btu/°F)
CFOODO	=	Thermal mass of outer food/inner walls node (Btu/°F)
CHILR	=	A logical variable which is TRUE if a self-contained refrigeration unit is present and turned on; otherwise, it is FALSE
CPG	=	Ambient gas specific heat (Btu/lb °F)
DELTA	=	Maximum temperature change of all nodes during a single internal iteration (°F)
DT	=	Temperature change computed for each node during a single internal iteration (°F)
DTH	=	System transient time step (hours)
DTYME	=	Internal transient time step used (hours)
FAN	=	A logical variable which is TRUE if there is a circulation fan present and turned on inside the food locker; otherwise, it is FALSE
HC	=	Heat transfer coefficient; air to food locker walls and food surface (Btu/hr-sq ft-°F)
LFLAG	=	Logic flag equal to 2 for steady-state case with fan off; otherwise, equal to -2
LL	=	Number of transient iterations required for thermal balance
LPASS	=	Counter for number of internal iterations of thermal balance equations
NFLAG	=	Flag used in logic to compute ambient gas properties
PWCAB	=	Water vapor partial pressure in cabin gas (psia)
PWF	=	Water vapor partial pressure inside locker (psia)
Q51	=	Total heat transferred into outer food node (Btu/hr)
Q78	=	Heat transferred from food inner to outer node (Btu/hr)
QCOOL	=	R(65) = Net cooling provided to locker (Btu/hr)

TABLE 3-4 (concluded)

DEFINITION OF CHILLR SUBROUTINE VARIABLES

QDOOR	=	Heat added during current iteration from opening door (Btu/hr)
QFAN	=	Heat from food locker fan which is dissipated within the fan housing (Btu/hr)
QFANA	=	Heat from food locker fan work of compression of the gas within the locker (Btu/hr)
QFANF	=	Heat transfer from fan housing to locker inner walls and food outer surface (Btu/hr)
QFREEZ	=	Latent heat required to freeze all the liquid water in food inner or outer nodes (Btu)
QLAT	=	Latent heat of moisture (condensation and/or fusion) in air entering locker door (Btu/lb)
QREJ	=	Heat rejected by self-contained cooling unit to cabin gas (Btu/hr)
R51	=	Variable used for temporary storage of R(51) during thermal balance
R98F	=	Absolute humidity of gas inside locker (lb water vapor/lb dry air)
RHOG	=	Ambient gas density (lb/cu ft)
SG2	=	Effective sum of thermal conductors to cooling coils (or other cold sink); (Btu/hr-°F)
SG51	=	Effective sum of thermal conductors to outer food node (Btu/hr-°F)
SG75	=	Effective sum of thermal conductors to inner food node (Btu/hr-°F)
TC	=	Temperature of cooling coils (or other cold sink); (°F)
TG	=	Ambient gas temperature (°F)
X - γ	=	Multiplier in steady-state heat balance equation (3.1.13)
α	=	Difference in absolute humidity between ambient air and air inside chiller locker (lb water vapor/lb dry air)

3.2 FTRAY

The FTRAY subroutine is designated as G-189A No. 66.

3.2.1 Subroutine Description

This subroutine simulates the performance of a Skylab-type food warming/serving tray. A typical food tray, shown in Figure 3-4, had eight recessed food cavities, of which three had embedded thermostatically controlled electrical resistance heaters to warm the food. The thermal model used for a single food warming cavity is shown in Figure 3-5. The cavity is assumed cylindrical and is subdivided into five food nodes of equal volume. In addition to the thermal network shown in Figure 3-5, the food is thermally connected to the ambient surroundings using the G-189A subroutine QSURR.

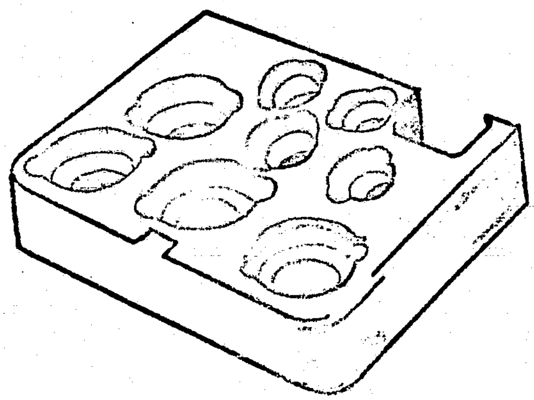


Figure 3-4. Typical Skylab-type Food Warming/Serving Tray

3.2.2 Subroutine Data

3.2.2.1 General Notes

a. Cabin air flow over the food tray component is optional. If an upstream component is used to provide air flow, it must be connected to the food tray primary side. All flow codes are allowed.

b. The total heat lost from all food trays to the ambient surroundings is given in R(53). This is the sum of convection [R(56)], radiation [R(59)], and conduction [R(62)]. If air flow is provided by an upstream component, the convective heat [R(56)] is transferred to that flow. All heat loss not rejected to the primary side air flow should be added to the cabin sensible heat load [cabin component R-array location R(66)] in GPOLY1.

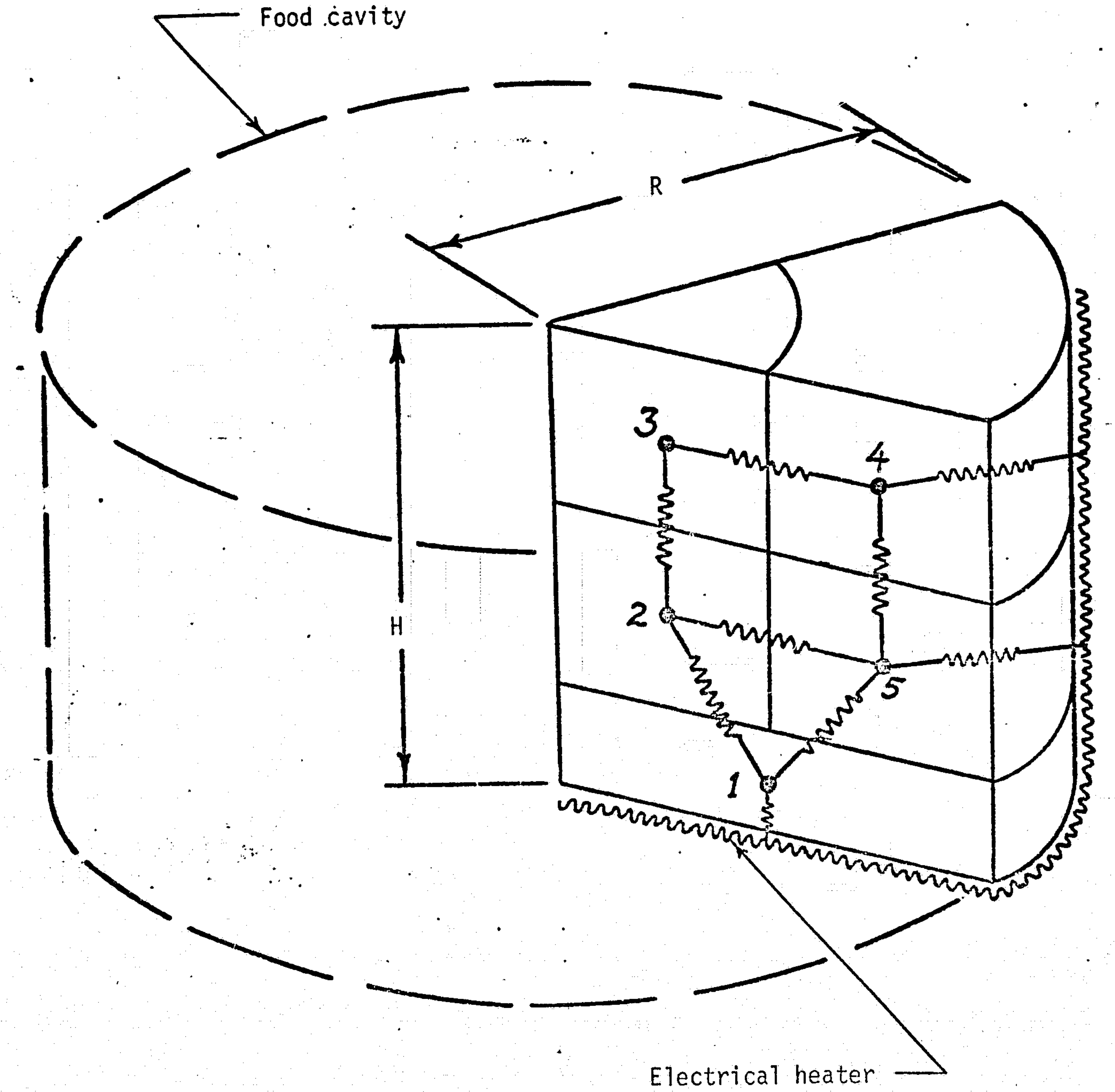


Figure 3-5. Thermal Model of a Single Food Warming Cavity

3.2.2.1 (Continued)

c. Any number of heated food cavities may be simulated by a single G-189A component using FTRAY. The routine determines the performance of a single food cavity and assumes all others are identical. If some food trays have different input data or time schedules, they must be simulated by separate G-189A components.

3.2.2.2 Instruction Options

NSTR(1): Initial conditions for the five food nodes

= 0 Assume properties for all five nodes initially equal, and input temperature and water content for node 1 only in R(71) and R(96)

= 1 Input initial temperature and water content for all five nodes in R(71) through R(75) and R(96) through R(100)

3.2.2.3 Heat Loss V-Array Data

Reference Location	Description	Data Type
51	Average temperature of the five food nodes (°F)	0
52	Summed thermal conductance from one food cavity to ambient surroundings (Btu/hr °F)	0
53	Total heat loss from all heated food cavities to surroundings (Btu/hr)	0
54	Ambient gas temperature (°F)	I(0)
55	Thermal convection conductor from one food cavity to ambient gas (Btu/hr °F)	I(0)
56	Convective heat loss from all heated food cavities to ambient gas (Btu/hr)	0
57	Ambient wall temperature (°F)	I(0)

3.2.2.3 (Continued)

Reference Location	Description	Data Type
58	Thermal radiation conductor (A3) between one food cavity and ambient walls (sq ft)	I(0)
59	Radiative heat loss from all heated food cavities to ambient walls (Btu/hr)	0
60	Temperature of ambient structure connected to food node (°F)	I(0)
61	Thermal conductor from one food node to attached ambient structure (Btu/hr °F)	I(0)
62	Thermal heat loss from all heated food cavities to attached ambient structure (Btu/hr)	0

3.2.2.4 Steady-State K-Array Data

Reference Location	Description	Data Type
16	Not used	
17	Not used	
18	Ice/water transition indicator. 0 = water in all food nodes is frozen. 1 = some food nodes are in ice/water transition. 2 = water in all food nodes is liquid.	0

3.2.2.5 Steady-State V-Array Data

Reference Location	Description	Data Type
65	Total heater power for all heated food cavities (Btu/hr)	0
66	Number of food cavities to be heated	I(R)
67	Single food cavity radius (in).	I(0)

3.2.2.5 (Continued)

Reference Location	Description	Data Type
68	Single food cavity depth (in).	I(0)
69	Single food cavity average heater power (watts)	I(0)
70	Maximum sum of thermal conductors to any one food node (Btu/hr °F)	0
71	Temperature at food node 1 (°F)	I(0), 0
72 73 74 75	Temperature at food node 2 (°F) Temperature at food node 3 (°F) Temperature at food node 4 (°F) Temperature at food node 5 (°F)	I(R) if NSTR(1)=1; otherwise, these are automatically set equal to R(71), 0
76	Thermal conductor from food node 3 to 4 (Btu/hr °F)	0
77	Thermal conductor from food node 2 to 3 (Btu/hr °F)	0
78	Thermal conductor from food node 1 to 2 (Btu/hr °F)	0
79	Thermal conductor from food node 1 to 5 (Btu/hr °F)	0
80	Fraction of heater power on food node 4 (or 5)	0
81	Fraction of heater power on food node 1	0
82	Frozen food thermal conductivity (Btu/hr-ft-°F)	I(0)
83	Unfrozen food thermal conductivity (Btu/hr-ft-°F)	I(0)

3.2.2.6 Steady State K-Array Data Used for Transient Case

Reference Location	Description	Data Type
16	Food node number to be used for automatic temperature control, 1 through 5	I(0)
17	This parameter stores the heater ON/OFF mode on previous time step, and is used for automatic temperature control. 0=OFF, 1=ON	0
18	Ice/water transition indicator. 0 = water in all food nodes is frozen. 1 = some food nodes are in ice/water transition. 2 = water in all food nodes is liquid.	0

3.2.2.7 Transient V-Array Data

Reference Location	Description	Data Type
65	Total average heater power for all heated food cavities during previous time step (Btu/hr)	0
66-68	Same as for steady state	
69	Not used	
70-79	Same as for steady state	
80	Fraction of heater power and thermal mass assigned to food node 4 (or 5)	0
81	Fraction of heater power and thermal mass assigned to food node 1	0
82-83	Same as for steady state	
84	Heater control temperature (°F)	I(0)
85	Heater control temperature dead band (°F) [$T_{control}$=R(84) ± R(85)]	I(0)

3.2.2.7 (Continued)

Reference Location	Description	Data Type
86	Specific heat of unfrozen food (Btu/lb wet food-°F)	I(0)
87	Specific heat of bone-dry food (Btu/lb dry food-°F)	0
88	Effective thermal mass of food heating tray connected to a single heating cavity (Btu/°F)	I(0)
89	Moisture content of food (lb H_2O/lb total wet food weight)	I(0)
90	Maximum heater power for a single heating cavity (watts)	I(0)
91	Thermal mass of food node 1 (Btu/°F)	0
92	Thermal mass of food node 2 (Btu/°F)	0
93	Thermal mass of food node 3 (Btu/°F)	0
94	Thermal mass of food node 4 (Btu/°F)	0
95	Thermal mass of food node 5 (Btu/°F)	0
96	Fraction of moisture in food node 1 which is frozen	I(R) only if initial food temp. is 32.0°F,0
97	Fraction of moisture in food node 2 which is frozen	
98	Fraction of moisture in food node 3 which is frozen	
99	Fraction of moisture in food node 4 which is frozen	
100	Fraction of moisture in food node 5 which is frozen	
101	Food mass density (lb/cu ft)	I(0)
102	Dry food mass in a single food node (lbs)	0
103	Mass of moisture in a single food node (lbs)	0

3.2.2.7 (Continued)

Reference Location	Description	Data Type
104	Food tray ON-OFF switch. Set to -1.0 if tray is turned off, and to +1.0 if tray is in a warming cycle. Note: Power to tray may be internally turned off, even when this switch is on, due to heater thermostat controller. Each time tray is turned on, reinitialize food temperature according to NSTR(1).	I(0)
105	Stores value of R(104) on previous iteration; used for ON/OFF initialization logic.	0

3.2.3 Analytical Model Description

3.2.3.1 Geometric Considerations

A single cylindrical food heating cavity is divided into five food nodes, as shown in Figure 3-5. Based on the assumption that all five nodes have the same volume, the following parameters based on geometry may be computed (see Table 3-7 for definition of symbols):

Volume

$$V_1 = V_2 = V_3 = V_4 = V_5 = 0.2\,\pi R^2 H \tag{3.2.1}$$

Height

$$h_1 = 0.2\ H \tag{3.2.2}$$

$$h_2 = h_3 = h_4 = h_5 = 0.4\ H \tag{3.2.3}$$

3.2.3.1 (Continued)

Heater Areas

$$A_{H_T} = \pi R^2 + 2 \pi RH \tag{3.2.4}$$

$$A_{H_2} = A_{H_3} = 0 \tag{3.2.5}$$

$$A_{H_4} = A_{H_5} = 0.8 \pi RH \tag{3.2.6}$$

$$A_{H_1} = \pi R^2 + 0.4 \pi RH \tag{3.2.7}$$

Outer Node Radii

$$r_1 = r_4 = r_5 = R \tag{3.2.8}$$

$$r_2 = r_3 = 0.707 R \tag{3.2.9}$$

Radii of Node Centers (placed at geometric center of node volume)

$$r_1^* = 0.707 R \tag{3.2.10}$$

$$r_2^* = r_3^* = 0.5 R \tag{3.2.11}$$

$$r_4^* = r_5^* = 0.866 R \tag{3.2.12}$$

Length Between Node Centers

$$L_{34} = L_{25} = 0.366 R \tag{3.2.13}$$

3.2.3.1 (Continued)

$$L_{23} = L_{45} = 0.4\ H \tag{3.2.14}$$

$$L_{12} = \sqrt{0.09\ H^2 + 0.04285\ R^2} \tag{3.2.15}$$

$$L_{15} = \sqrt{0.09\ H^2 + 0.02528\ R^2} \tag{3.2.16}$$

"Window" Area Between Nodes

$$A_{34} = A_{25} = 0.5656\ \pi\ RH \tag{3.2.17}$$

$$A_{23} = A_{45} = A_{12} = A_{15} = 0.5\ \pi\ R^2 \tag{3.2.18}$$

Heater Flux ($\mathcal{H} \equiv$ total heater power)

$$\text{Total Heater Area} = 2\ \pi\ RH + \pi\ R^2$$

$$q_{H_4} = q_{H_5} = \left[\frac{1}{2.5 + 1.25\ \frac{R}{H}}\right]\mathcal{H} \tag{3.2.19}$$

$$q_{H_1} = \left[\frac{0.4 + \frac{R}{H}}{2 + \frac{R}{H}}\right]\mathcal{H} \tag{3.2.20}$$

Thermal Conductors (k ≡ food thermal conductivity)

$$G_{34} = G_{25} = 1.548\ \pi\ Hk \tag{3.2.21}$$

$$G_{23} = G_{45} = 1.25\ \pi\ \frac{R^2}{H}k \tag{3.2.22}$$

3.2.3.1 (Continued)

$$G_{12} = \frac{0.5\ \pi\ R^2 k}{\sqrt{0.09\ H^2 + 0.0429\ R^2}} \qquad (3.2.23)$$

$$G_{15} = \frac{0.5\ \pi\ R^2 k}{\sqrt{0.09\ H^2 + 0.0253\ R^2}} \qquad (3.2.24)$$

3.2.3.2 Thermal Balance

The thermal solution is divided into two parts: that within the food cavity, and thermal exchange between the food and ambient surroundings. These two solutions are interdependent since:

a. The solution within the food cavity provides the average food temperature for computing ambient heat loss.

b. The thermal balance with ambient surroundings provides the boundary conditions for heat transfer within the food cavity.

The heat loss from the food to ambient surroundings is found first using the G-189A subroutine QSURR. This routine determines the total conductive, radiative, and convective thermal loss from the food. The arithmetic average temperature of the five food nodes is used as the effective base temperature in the subroutine. The top food surface is assumed to be covered with a lid, as on Skylab, with negligible moisture evaporation. Thus, the boundary conditions for the food cavity are:

a. The heat lost at the heater surface equals the total conductive heat (q_{cond}) from the food to surroundings;

b. The heat lost at the top surface equals the total radiative (q_r) and convective (q_{conv}) heat from the food to surroundings.

3.2.3.2 (Continued)

The thermal balance within the food cavity may then be made by applying the following standard equations at each food node:

$$\sum_{i=1}^{5} q_i = m\, c_{p_i} \frac{\Delta T_i}{\Delta t} \qquad (3.2.25)$$

for a transient solution, and

$$\sum_{i=1}^{5} q_i = 0 \qquad (3.2.26)$$

for a steady-state solution. The net heat q_i into each food node is given in terms of the nodal surface areas (equations 3.2.17 and 3.2.18), heater areas (equations 3.2.5 and 3.2.6), heater flux (equations 3.2.18 and 3.2.19), and thermal conductors (equations 3.2.20 through 3.2.23), as follows:

$$\sum q_1 = q_{H_1} - \left(\frac{A_{H_1}}{A_{H_T}}\right) q_{cond} + G_{12}\,(T_2 - T_1) + G_{15}\,(T_5 - T_1) \qquad (3.2.27)$$

$$\sum q_2 = G_{12}\,(T_1 - T_2) + G_{23}\,(T_3 - T_2) + G_{25}\,(T_5 - T_2) \qquad (3.2.28)$$

$$\sum q_3 = -\tfrac{1}{2}\,(q_r + q_{conv}) + G_{23}\,(T_2 - T_3) + G_{34}\,(T_4 - T_3) \qquad (3.2.29)$$

$$\sum q_4 = q_{H_4} - \left(\frac{A_{H_4}}{A_{H_T}}\right) q_{cond} - \tfrac{1}{2}\,(q_r + q_{conv}) + G_{34}\,(T_3 - T_4) + G_{45}\,(T_5 - T_4) \qquad (3.2.30)$$

$$\sum q_5 = q_{H_5} - \left(\frac{A_{H_5}}{A_{H_T}}\right) q_{cond} + G_{15}\,(T_1 - T_5) + G_{25}\,(T_2 - T_5) + G_{45}\,(T_4 - T_5) \qquad (3.2.31)$$

3.2.3.2 (Continued)

Equations (3.2.25) through (3.2.31) are solved each time step using a standard forward-difference method as follows:

$$T_{i_{new}} = T_{i_{old}} + \gamma \sum q_i \tag{3.2.32}$$

For steady-state solutions, the value used for γ is given by

$$\gamma_{steady\ state} = \frac{0.8}{\text{maximum sum of conductors to ground}} \tag{3.2.33}$$

Internal steady-state iterations are made using equations (3.2.32) and (3.2.33) until the old and new temperatures differ by 0.05°F or until 25 iterations have been made.

For transient solutions, the value for γ in equation (3.2.32) is

$$\gamma_{transient} = \frac{\Delta t}{m\ c_{p_i}} \tag{3.2.34}$$

The thermal mass $m\ c_p$ of the node will vary with the ice/water balance, as discussed further in the following section. The computing time increment Δt is first computed as 0.7 times the smallest of the time increments required for computational stability, where

$$\Delta t_{stability} = \frac{\text{thermal mass}}{\text{sum of conductors to ground}} \tag{3.2.35}$$

The time increment is further limited to the system computing time step. In addition, it was found that large time steps could cause the automatic heater control to leave the heater on too long; thus, causing excessive temperature fluctuations in the food nodes adjacent to the heater. Therefore, as suggested in Reference 4, the internal time increment was further

3.2.3.2 (Continued)

limited to 20 seconds. If necessary, internal iterations are performed until the required system time step is reached.

3.2.3.3 Thawing Effects

Some thermal properties for typical foods, taken from Reference 4, are given in Table 3-6. It is apparent that some properties are highly dependent on the physical state (whether liquid or ice) of the food moisture. Thus, the thermal mass for each node is computed separately:

$$m\,c_{p_i} = (m_{dry\ food}\,c_{p_{dry\ food}}) + m_{liquid\ water} + 0.46\ m_{ice} \tag{3.2.36}$$

As a frozen food node reaches 32°F, its temperature is held constant and the heat input assumed to thaw its moisture. The water/ice balance during this transition is computed for evaluating thermal mass from equation (3.2.36). The food thermal conductivity is assumed to be that for frozen food at all nodes until all the water is melted. Then the conductivity for thawed food is used.

3.2.3.4 Heater Control

For a transient solution, the heater power is controlled automatically to an input set point temperature. Any node may be used as a temperature sensor, but normally node 1 should be used since it will be hottest. A control deadband is also used. Since the internal computing time increment may be less than the system time step, as discussed in Paragraph 3.2.3.2, the heater may be cycled on and off several times during one transient time step. The integrated average heater power during one time step is computed and output in R-array location R(65). For a steady-state solution, only the average heater power should be input, and the heater assumed to be continually energized.

3.2.3.5 Pressure Drop Considerations

The food warming/serving trays are normally exposed only to the cabin air convection currents and are not physically connected in an actual fluid flow loop. Thus, no pressure drop calculations would be required for this component.

TABLE 3-5

DEFAULT VALUES FOR FTRAY SUBROUTINE INPUT DATA

K(16)=1

R-Array* Location	Default Value	
54	70	
55	0.148	
57	70	
58	0.0625	
60	70	
61	0.216	
67	1.875	
68	1.125	
69	7.9	
71	140	steady state
	- 10	transient, R(104) > 0
	70	transient, R(104) < 0
82	0.99	
83	0.25	
84	157	
85	2	
86	0.78	
88	1.9	
89	0.724	
90	52	
101	59	
104	1	

*See Section 3.2.2 for definition of variables.

TABLE 3-6

THERMAL PROPERTIES OF SELECTED SKYLAB FOODS

Food Substance	Water Content (%)	Density $\frac{lb_m}{ft^3}$	Latent Heat of Fusion $\frac{Btu}{lb_m}$	Specific Heat (Btu/lb_m-F) **	Specific Heat (Btu/lb_m-F) †	Thermal Conductivity (Btu/hr-ft-F) **	Thermal Conductivity (Btu/hr-ft-F) †	Thermal Diffusivity (ft^2/hr) **	Thermal Diffusivity (ft^2/hr) †
Pre-buttered Roll	23.8*	15.*	34.	0.70	0.34	0.19	0.45	0.018	0.088
Coffee Cake	25.0*	15.*	36.	0.70	0.34	0.19	0.46	0.018	0.088
Filet Mignon	63.2*	58.	91.	0.71	0.40	0.26	0.97	0.0063	0.042
Chili with Meat	66.9*	58.	96.	0.74	0.40	0.26	0.99	0.0061	0.043
Prime Rib of Beef	72.4*	59.	104.	0.78	0.42	0.27	1.04	0.0059	0.042
Lobster Newberg	75.5*	59.	109.	0.80	0.43	0.28	1.09	0.0059	0.043
Stewed Tomatoes	88.4	61.	129.	0.91	0.47	0.30	1.25	0.0054	0.044

*Experimentally determined for actual Skylab foods
**Unfrozen
†Frozen

TABLE 3-7

DEFINITION OF SYMBOLS FOR FTRAY SUBROUTINE DESCRIPTION

A_{H_i} = Area of heater contact with food node "i" (sq ft)

A_{H_T} = Total heater surface area (sq ft)

A_{ij} = "Window" cross-sectional area between adjacent nodes "i" and "j" (sq ft)

c_{p_i} = Thermal specific heat of node "i" (Btu/lb °F)

G_{ij} = Thermal conductor between adjacent food nodes "i" and "j" (Btu/hr °F)

H = Food cavity height (ft)

$\mathcal{H}$ = Total heater input electrical power (Btu/hr)

h_i = Height of food node "i" (ft)

k = Food thermal conductivity (Btu/hr-ft-°F)

L_{ij} = Length between centers of adjacent nodes "i" and "j" (ft)

m = Total mass of food node (lb)

q_{cond} = Heat lost by conduction from food to ambient surroundings (Btu/hr)

q_{conv} = Heat lost by convection from food to ambient surroundings (Btu/hr)

q_{H_i} = Heater energy input into node "i" (Btu/hr)

q_i = Heat flow into node "i" (Btu/hr)

q_r = Heat lost by radiation from food to ambient surroundings (Btu/hr)

R = Food cavity radius (ft)

r_i = Outer radius of food node "i" (ft)

r_i^* = Radius of center of food node "i" (ft)

TABLE 3-7 (concluded)

DEFINITION OF SYMBOLS FOR FTRAY SUBROUTINE DESCRIPTION

T_i	= Temperature at node "i" (°F)
Δt	= Computing time increment (hrs)
V_i	= Volume of food node "i" (cu ft)
γ	= Multiplier to compute new temperatures, defined by equation (3.2.32)

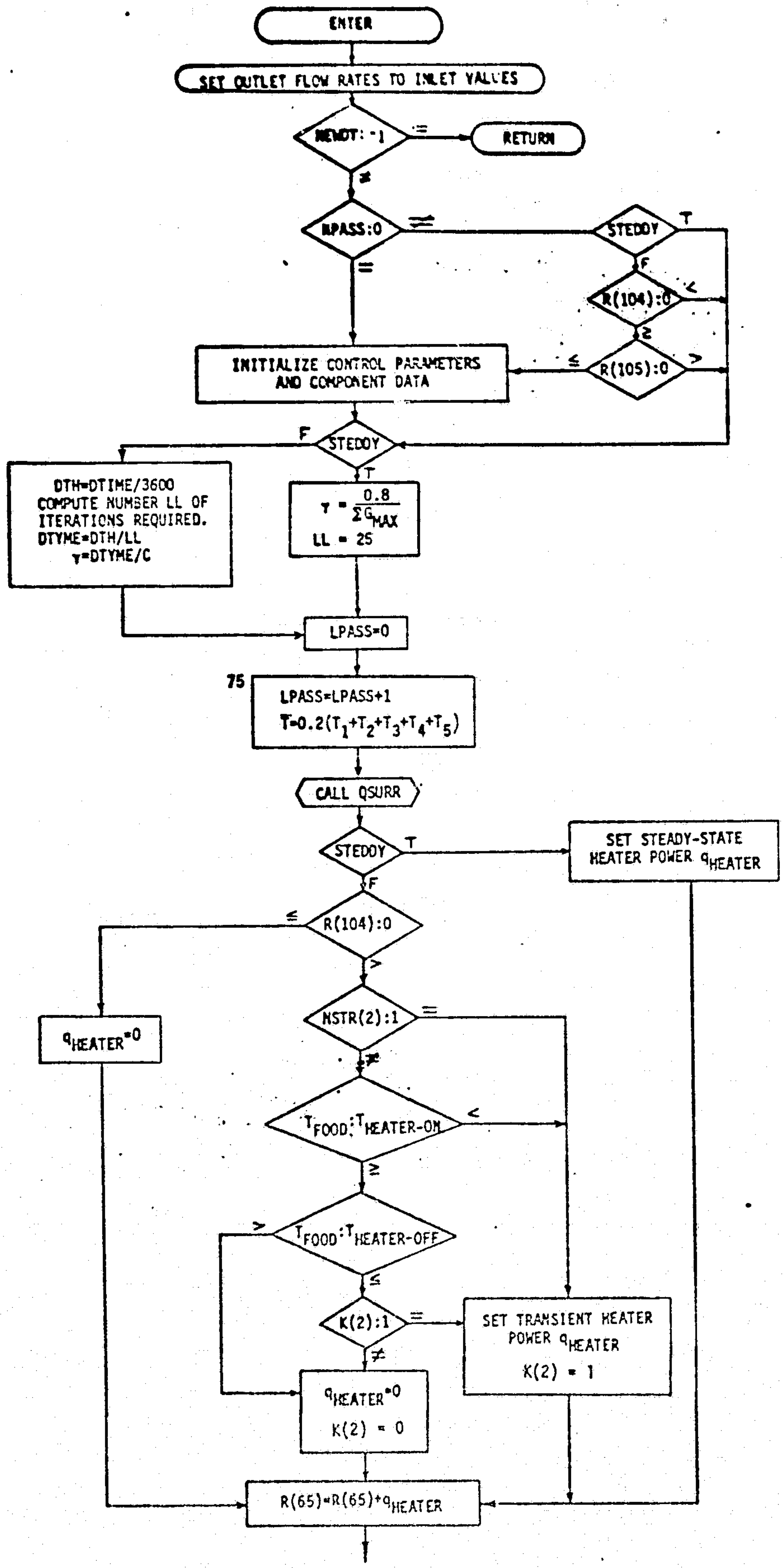


Figure 3-6. Logic Flow Chart for FTRAY Subroutine

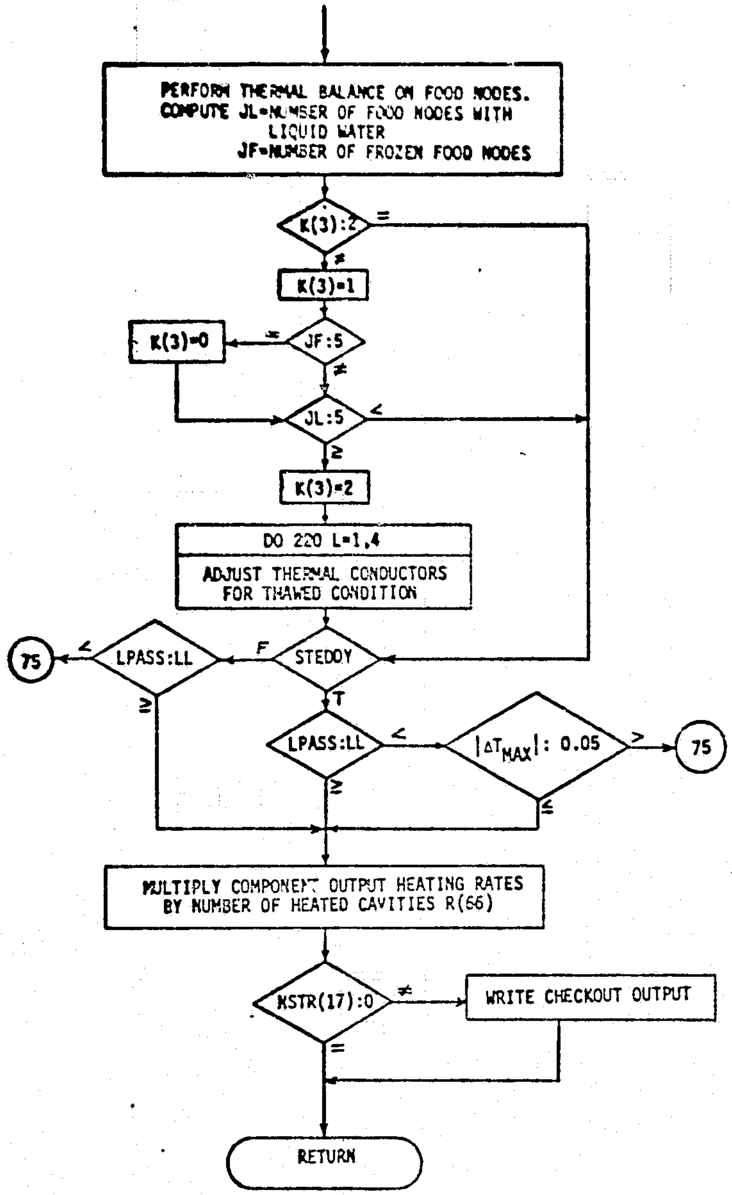


Figure 3-6. Logic Flow Chart for FTRAY Subroutine (concluded)

TABLE 3-8

DEFINITION OF FTRAY SUBROUTINE VARIABLES

ASP = Food cavity aspect ratio, radius/height (dimensionless)

COND = Food effective thermal conductivity (Btu/hr-ft-°F)

DELTA = Maximum temperature change of all nodes during a single internal iteration (°F)

DT = Temperature change computed for each food node during a single internal iteration (°F)

DTH = System transient time step (hours)

DTYME = Internal transient time step used (hours)

GMAX = Maximum sum of thermal conductors to any food node (Btu/hr °F)

H = Food cavity height (ft)

JF = Number of food nodes with all water frozen

JL = Number of food nodes with all water liquid

LL = Maximum number of internal iterations of heat balance equation

LPASS = Counter for number of iterations of thermal balance equations

Q(1),Q(2),Q(3),Q(4),Q(5) = Net heat transferred into food nodes 1, 2, 3, 4, and 5 (Btu/hr)

QA3 = Heat transferred from ambient air into food node 3 (Btu/hr)

QHTR = Net heat into food cavity from electrical heater (less the heat conducted into the tray); (Btu/hr)

Q_{ij} = Heat transfer from food node "1" to "j" (Btu/hr)

QMELT = Latent heat required to melt the ice in each food node (Btu)

QW4 = Heat transferred from heater wall into food node 4 (Btu/hr)

RAD = Food cavity radius (ft)

SG1,SG2,SG5 = Sum of thermal conductors to food nodes 1, 2, and 5 (Btu/hr-°F)

T1,T2,T3,T4,T5 = Temperature of food nodes 1, 2, 3, 4, and 5 (°F)

TAVG = Arithmetic average of the five food node temperatures (°F)

TABLE 3-8 (concluded)

DEFINITION OF FTRAY SUBROUTINE VARIABLES.

WDRY = Weight of dry food only in a single food node (lb)

WH2O = Weight of water in a single food node (lb)

X = γ = Multiplier in heat balance equation (3.2.32)

3.3 ROSMOS

The ROSMOS subroutine is designated as G-189A No. 69.

3.3.1 Subroutine Description

Reverse osmosis is a process used for removing impurities from waste water. Fluid is forced by static pressures across a membrane in a direction opposite to which it would normally flow due to osmotic pressure alone. Most of the impurities are filtered out by the membrane, thus leaving a purified water flow through the unit. This technique is in use commercially in many water and waste treatment applications and is being developed to recycle spacecraft waste water. The subroutine will simulate the performance of an arbitrary type of reverse osmosis unit, shown in Figure 3-7, if test or design performance data for that unit are available. Only a single-stage unit may be simulated by a single component, but multistage operation may be achieved by connecting more than one unit together.

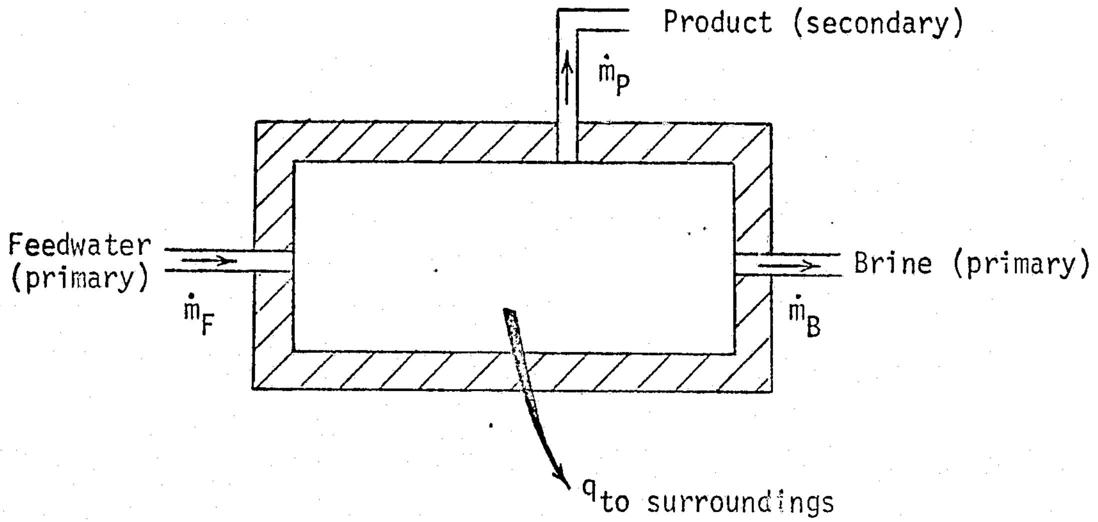


Figure 3-7. ROSMOS Component Flow Schematic

3.3.2 Subroutine Data

3.3.2.1 General Notes

a. Feedwater must be inlet on the primary side. Treated product water will be outlet on the secondary side, and the brine outlet at the primary side.

3.3.2.1 (Continued)

b. Only liquid flow codes (0 or 4) may be used. Results for flow code 4 will only be correct if the recovery factors for each special flow type are the same.

c. The reverse osmosis unit is unique among G-189A components in that the secondary (product) outlet pressure must be calculated every time to obtain a solution, even when normal pressure drop calculations are not being made. To do this, the user must input the value to be used for product water pressure in GPOLY logic in R(74).

3.3.2.2 Heat Loss V-Array Data

Reference Location	Description	Data Type
51	Average reverse osmosis unit core temperature (°F)	I(0), 0
52	Summed thermal conductance between reverse osmosis unit and ambient - excluding conductance to liquid flow through unit (Btu/hr °F)	0
53	Reverse osmosis unit net heat loss to ambient - excluding heat transfer to liquid flow through unit (Btu/hr)	0
54	Ambient gas temperature (°F)	I(0)
55	Thermal convection conductor from reverse osmosis unit outer surface to ambient gas (Btu/hr °F)	I(R)
56	Convective heat loss from reverse osmosis unit to ambient gas (Btu/hr)	0
57	Ambient wall temperature (°F)	I(0)
58	Thermal radiation conductor (A℈) between ambient walls and reverse osmosis unit surface (sq ft)	I(R)
59	Radiative heat loss from reverse osmosis unit to ambient walls (Btu/hr)	0

3.3.2.2 (Continued)

Reference Location	Description	Data Type
60	Temperature of ambient structure connected to reverse osmosis unit (°F)	I(0)
61	Thermal conductor from reverse osmosis unit to attached ambient structure (Btu/hr °F)	I(R)
62	Thermal heat loss from reverse osmosis unit to attached ambient structure (Btu/hr)	0
63	Reverse osmosis unit external surface temperature (°F)	0
64	Thermal conductor through insulation around reverse osmosis unit (Btu/hr °F); may be zero if unit is uninsulated	I(R)

3.3.2.3 Steady-State K-Array Data

Reference Location	Description	Data Type
16	Number of individual types of salts to be handled by reverse osmosis unit (extra V-array locations must be five times this value)	I(0)

3.3.2.4 Steady-State V-Array Data

Reference Location	Description	Data Type
65	Overall design water recovery factor, $\dot{m}_{product}/\dot{m}_{feed}$ (dimensionless)	I(0)
66	Overall design total salt rejection factor	I(0)
67	Design product water pressure (psia)	I(0)
68	Brine overall total salt concentration (lb salt/lb water)	0

3.3.2.4 (Continued)

Reference Location	Description	Data Type
69	Product water overall total salt concentration (lb salt/lb water)	0
70	Design feedwater flow rate (lb/hr)	I(0)
71	Design feedwater overall total salt concentration (lb salt/lb water)	I(0)
72	Actual feedwater overall total salt concentration (lb salt/lb water)	I(0)
73	Design feedwater pressure (psia)	I(0)
74	Actual product pressure (psia) input in GPOLY1 from downstream component	I(0)
75	Overall design average osmotic pressure (psia)	I(0)

3.3.2.5 Transient V-Array Data

Reference Location	Description	Data Type
76	Thermal capacitance of reverse osmosis unit-dry (Btu/°F)	I(0)

3.3.2.6 Extra V-Array Data [need 5J locations where J=K(16)]

Reference Location	Description	Data Type
77→76 + J	Design concentration of individual feedwater salts (lb salt/lb water)	I(0)
77 + J to 76 + 2J	Actual concentration of individual feedwater salts (lb salt/lb water)	I(0)
77 + 2J to 76 + 3J	Design individual salt rejection factor (dimensionless)	I(0)

3.3.2.6 (Continued)

Reference Location	Description	Data Type
77 + 3J to 76 + 4J	Actual concentration of individual salts in product water (lb salt/lb water)	0
77 + 4J to 76 + 5J	Actual concentration of individual salts in brine (lb salt/lb water)	0

3.3.3 Analytical Model Description

In a reverse osmosis unit, fluid is forced by static pressures across a membrane in a direction opposite to which it would normally flow due to osmotic pressure alone. This technique is in use commercially in many water and waste treatment applications and is being developed to recycle spacecraft waste water. Since a particular spacecraft reverse osmosis unit design has not been determined, the subroutine was written in general terms to handle any unit for which some design or test data are available. The input design data are adjusted for the off-design conditions present during a run. The thermal balance of the reverse osmosis unit with ambient surroundings is handled using the standard G-189A subroutine QSURR.

3.3.3.1 Flow Balance

The flow balance across a reverse osmosis module may be analyzed from two viewpoints of interest. First, one can consider the overall net effect of all the different types of solute impurities, lumped together and treated as a single homogeneous impurity. This approach is used in determining overall water balances and initial sizing and design of water recovery components. Secondly, one can consider the effect of the reverse osmosis module on each different type of impurity present in the water with its own individual rejection factor. This subroutine uses the former "overall" approach for handling the total flow balance, and will also apply the second approach as an option for handling any number of individual impurities desired.

3.3.3.1 (Continued)

The overall water recovery factor for a reverse osmosis unit is defined as

$$Y \equiv \frac{\dot{m}_P}{\dot{m}_F} \tag{3.3.1}$$

For conservation of mass across the unit,

$$\dot{m}_P + \dot{m}_B = \dot{m}_F \tag{3.3.2}$$

$$\dot{S}_P + \dot{S}_B = \dot{S}_F \tag{3.3.3}$$

where $\dot{m}$ refers to the total mass flow rate, and $\dot{S}$ is the effective overall mass flow rate of the solutes. The overall salt rejection factor for the unit is defined as

$$R \equiv \frac{\dot{S}_B}{\dot{S}_F} \tag{3.3.4}$$

Combining equations (3.3.1) through (3.3.4), the overall total salt concentration in the brine and product water is given as a function of the inlet conditions as follows:

$$C_B \equiv \frac{\dot{S}_B}{\dot{m}_B} = \frac{R\ C_F}{(1 - Y)} \tag{3.3.5}$$

$$C_P \equiv \frac{\dot{S}_P}{\dot{m}_P} = \frac{(1 - R)\ C_F}{Y} \tag{3.3.6}$$

3.3.3.1 (Continued)

Now the osmotic pressure of a given substance through a membrane has been found to vary with the concentration of that substance. It is assumed a proportional relationship exists such that

$$\Delta P_{osmosis} = \Delta P_{osmosis_{design}} \left(\frac{C_F}{C_{F_{design}}} \right) \tag{3.3.7}$$

The product water flow rate is assumed proportional to pressure drop across the membrane less the osmotic pressure; thus,

$$\dot{m}_P \propto P_F - P_P - \Delta P_{osmosis} \tag{3.3.8}$$

The design pressures and flow rates of the reverse osmosis unit are input. Therefore, the flow rate of the product water may be found from equation (3.3.8) as follows:

$$\dot{m}_P = \dot{m}_{P_{design}} \left[\frac{\left(P_F - P_P - \Delta P_{osmosis} \right)_{actual}}{\left(P_F - P_P - \Delta P_{osmosis} \right)_{design}} \right] \tag{3.3.9}$$

To use equation (3.3.9) to compute the product water total flow rate $\dot{m}_P$, the product pressure P_P must first be found. For an actual unit, this would be a function of the feedwater pressure, or it could be regulated by a downstream valve or pump. Since these data are not known, the user must supply the product water pressure in a GPOLY statement based on data for the actual unit being simulated. The design product flow $\dot{m}_{P_{design}}$ in equation (3.3.9) is found from equation (3.3.1):

$$\dot{m}_{P_{design}} = Y_{design} \, \dot{m}_{F_{design}} \tag{3.3.10}$$

3.3.3.1 (Continued)

It is assumed that the flow rate of salts in the product will be proportional to (1) the feedwater total salt concentration, and (2) the effective pressure drop across the membrane; that is,

$$\dot{S}_P \propto C_F \cdot (P_F - P_P - \Delta P_{osmosis}) \tag{3.3.11}$$

Combining equations (3.3.5), (3.3.9), and (3.3.11), the overall concentration of salts in the product water may be found:

$$C_P = C_{P_{design}} \left(\frac{C_F}{C_{F_{design}}} \right) \tag{3.3.12}$$

The design product concentration is found from equation (3.3.6) using input design values for R, C_F, and Y. Finally, the overall concentration of salts in the brine is found from equations (3.3.2) and (3.3.5):

$$C_B = \frac{C_F \dot{m}_F - C_P \dot{m}_P}{\dot{m}_B} \tag{3.3.13}$$

3.3.3.2 Individual Salt Concentration Balance

The overall flow equations in Paragraph 3.3.3.1 are sufficient to compute the outlet primary and secondary flows for the G-189A system. However, as an option, the subroutine will compute the product water and brine concentrations of any number of individual specific solutes. The same analysis in Paragraph 3.3.3.1 for the overall water/salt balance may be made for each individual salt with similar results. This would give, for the brine and product individual salt concentrations [see equations (3.3.12) and (3.3.13)]:

$$C_{i_P} = C_{i_{P_{design}}} \left(\frac{C_{i_F}}{C_{i_{F_{design}}}} \right); \; i=1, \text{ total number of salts} \tag{3.3.14}$$

3.3.3.2 (Continued)

$$C_{i_B} = \frac{C_{i_F}\dot{m}_F - C_{i_P}\dot{m}_P}{\dot{m}_B} \quad i=1, \text{ total number of salts} \tag{3.3.15}$$

Equations (3.3.14) and (3.3.15) will be evaluated in the subroutine for each individual solute requested by the user. For each solute, the design values (or test values if available) must be input for feedwater salt concentration C_{i_F} and salt rejection factor R_i. Also, the actual feedwater concentration of each solute must be computed in GPOLY logic and input. Again, the design product concentration is found from equation (3.3.6):

$$C_{i_{P_{design}}} = \frac{(1 - R_{i_{design}})C_{i_{F_{design}}}}{Y_{design}} \tag{3.3.16}$$

3.3.3.3 Thermal Balance

Thermal exchange between the reverse osmosis unit and ambient surroundings is accounted for using the standard G-189A subroutine QSURR. This model was included to accommodate certain designs requiring elevated temperatures, and simulates conduction, radiation, and convection from the core and insulation effects. The net heat into the single core node is given by

$$q_{in} = \dot{m}_F c_{P_F} (T_F - T_o) - q_{surr} \tag{3.3.17}$$

For the steady-state case, the net heat input must be zero; so the average core temperature is given by

$$T_{o_{steady\ state}} = T_F - \frac{q_{surr}}{\dot{m}_F c_{P_F}} \tag{3.3.18}$$

3.3.3.3 (Continued)

For a transient solution, the core temperature is found as follows:

$$T_{o_{new}} = T_{o_{old}} + \frac{\left[\dot{m}_F \, c_{p_F}\left(T_F - T_{o_{old}}\right) - q_{surr}\right] \Delta t}{m_o \, c_{p_o}} \qquad (3.3.19)$$

The computing time increment Δt is computed as 0.5 times the maximum time allowed for computational stability, where

$$\Delta t_{stability} = \frac{\text{core thermal mass}}{\text{sum of conductors to ground}} \qquad (3.3.20)$$

If the system time step is less than the stability time increment, then the system time step is used directly for Δt. If it is greater, then internal iterations of equation (3.3.19) are repeated until the system time step is reached.

3.3.3.4 Pressure Drop Considerations

The reverse osmosis unit has one flow inlet (primary feedwater) and two outlets (primary brine and secondary product water). This component is still in the design stage, and pressure drop model data cannot yet be specified. Typical designs now being considered have an approximate pressure range of 1000 psia feedwater, 50 psig drop to the brine, and 50 psia product water pressure. The feedwater flow rate may or may not be controlled by a downstream pressure regulator. If it were, the brine pressure should be set in GPOLY logic based on the regulator setting. This would not require use of the standard G-189A pressure drop options. If a regulator were not used, the G-189A option (1) would again probably be best to use for computing brine pressure (with the effective flow path area, length, and hydraulic diameter being input) unless actual experimental data were available.

3.3.3.4 (Continued)

The reverse osmosis unit is unique among G-189A components in that the product water (secondary outlet) pressure must be known in order to obtain a solution. Thus, it must be either input or calculated every time step whether a system pressure drop analysis is being made or not. Since the product is on the component secondary side, and there is no secondary inlet flow (feedwater is on the primary side), the standard G-189A pressure drop options may not be used to compute its pressure. Thus, the product pressure must be input or computed in GPOLY logic based on experimental data. In the Space Station system model described in Section 6, the product pressure is specified in R(74) as 45 psia. When the reverse osmosis unit design becomes finalized and pressure drop data are available, it is recommended they be incorporated into the ROSMOS subroutine; thus, eliminating this additional user input.

TABLE 3-9

DEFAULT VALUES FOR ROSMOS SUBROUTINE INPUT DATA

R-Array* Location	Default Value
51	75
54	70
57	70
60	70
65	0.96
66	0.93
67	50
70	9
71	0.01
72	0.008
73	900
74	45
75	200
76	2.5

*See Section 3.3.2 for definition of variables.

TABLE 3-10

DEFINITION OF SYMBOLS FOR ROSMOS SUBROUTINE DESCRIPTION

C = Solute concentration $\dot{S}/\dot{m}$ (lb solute/lb total fluid)

c_p = Thermal specific heat (Btu/lb °F)

$\dot{m}$ = Fluid mass flow rate (lb/hr)

P = Fluid pressure, psia

q_{surr} = Net heat loss from reverse osmosis unit to ambient surroundings (excluding heat transfer to the flowing fluid); (Btu/hr)

R = Overall total solute rejection factor defined by equation (3.3.4)

$\dot{S}$ = Effective overall solute mass flow rate (lb/hr)

T = Temperature (°F)

Y = Overall water recovery factor defined in equation (3.3.1)

Δt = Computing time increment (hours)

Subscripts

B = Brine

F = Feedwater

i = Individual solute

o = Reverse osmosis unit core

P = Product water

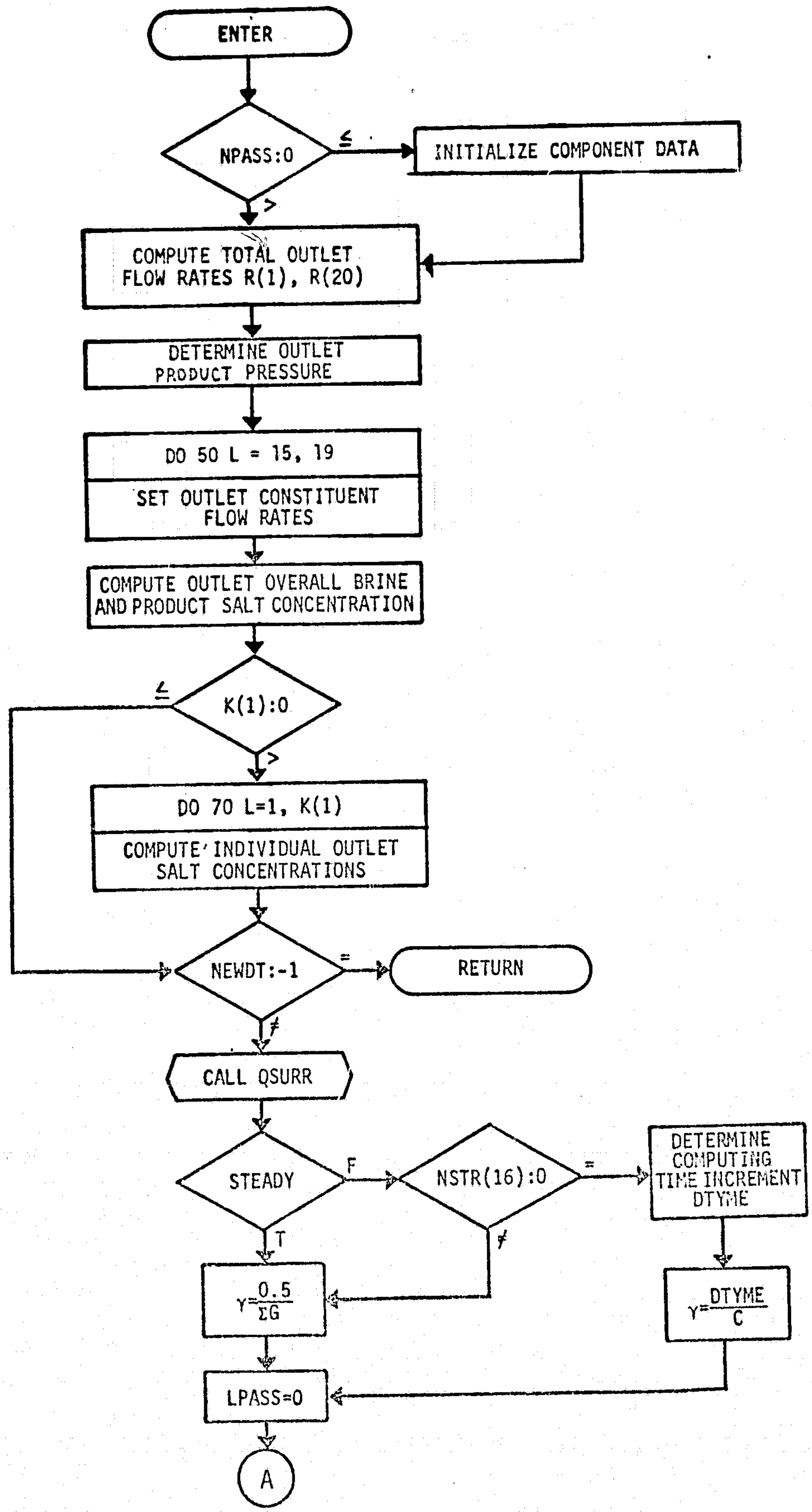


Figure 3-8. Logic Flow Chart for ROSMOS Subroutine

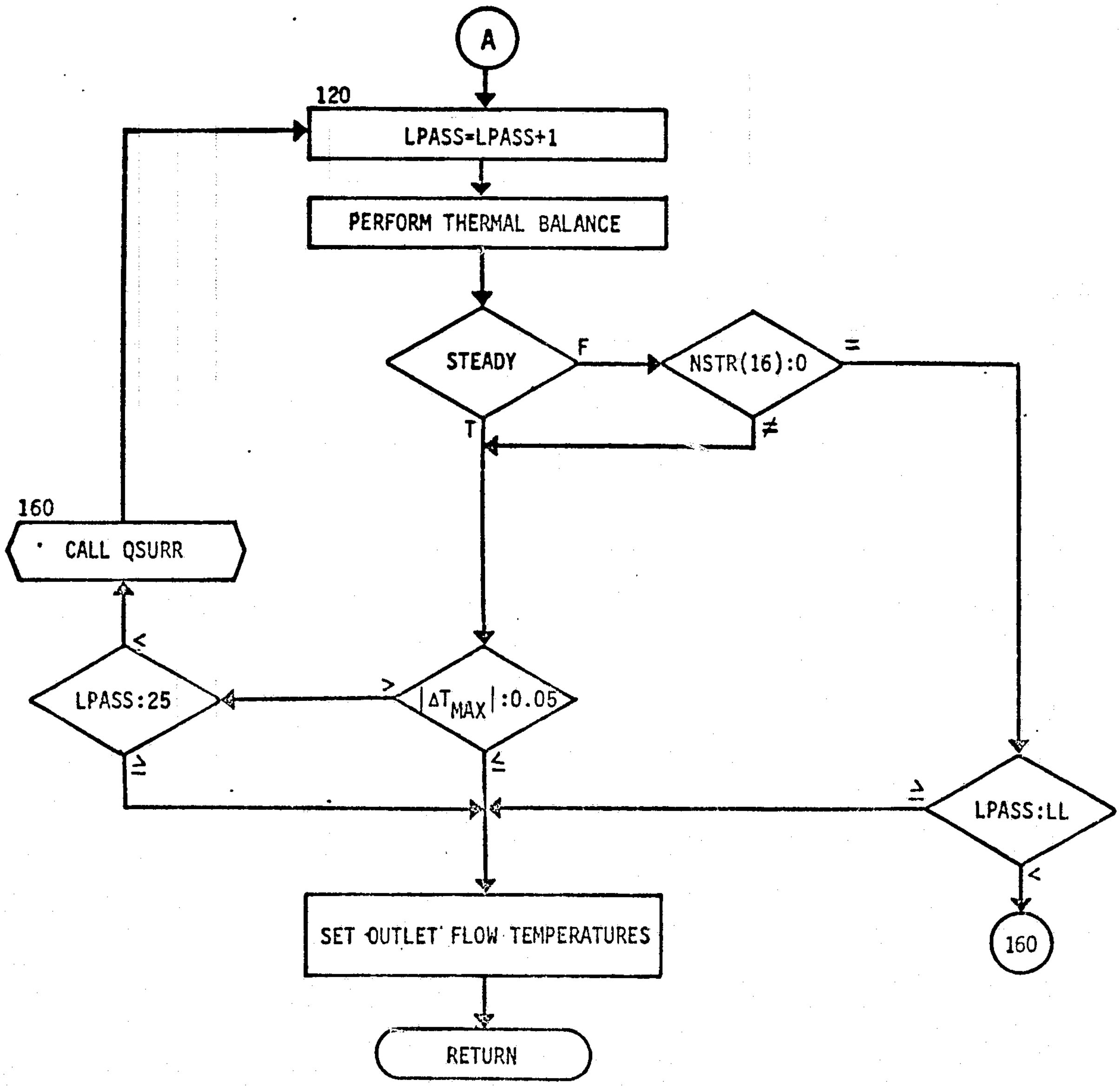


Figure 3-8. Logic Flow Chart for ROSMOS Subroutine (concluded)

TABLE 3-11

DEFINITION OF ROSMOS SUBROUTINE VARIABLES

CPRD = Concentration of solutes in product water under design (or test) conditions (dimensionless)

DTH = System transient time step (hours)

DTYME = Internal transient time step used (hours)

LL = Maximum number of internal iterations of heat balance equations

LPASS = Counter for number of iterations of heat balance equations

POSM = Effective osmotic pressure of salts across membrane (psi)

R51 = Variable used for temporary storage of R(51) during thermal balance

WPRD = Product mass flow rate under design (or test) conditions (lb/hr)

X = γ = Multiplier used in heat balance equation; net heat input times X equals required temperature change (°F/Btu/hr)

C-2

3.4 SHOWER

The SHOWER subroutine is designated as G-189A No. 67.

3.4.1 Subroutine Description

This subroutine models the thermal and evaporative mass exchange in a shower stall, as shown in Figure 3-9. The G-189A subroutine QSURR is used to model the thermal exchange between the shower stall frame and ambient environment. Evaporation is modeled by mass transfer equations as a function of air inlet flow rate, humidity and properties. The effect of the shower occupant is included by standard metabolic equations based on input metabolic rate and respiratory quotient.

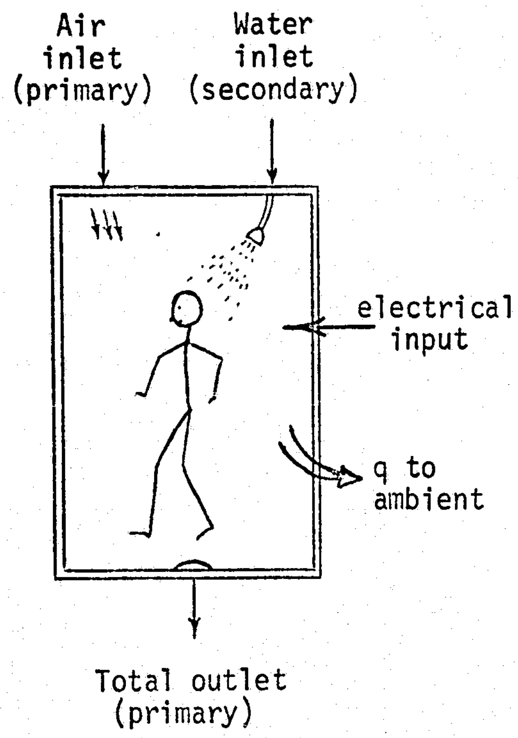


Figure 3-9. Shower Model Flow Schematic

3.4.2 Subroutine Data

3.4.2.1 General Notes

a. Air inlet is on the primary side; water inlet is on the secondary. Total outlet flow is on the primary side.

b. All gaseous flow codes are allowed (1, 2, or 3) on the primary side, and all liquid flow codes (0 or 4) on the secondary.

c. Oxygen and CO_2 balance will only be made for primary flow codes 2 and 3.

d. The shower stall is considered to be occupied if and only if there is flow inlet to the primary (air) side.

3.4.2.1 (Continued)

e. The thermal, metabolic oxygen, and CO_2 effects of the crewman are accounted for within the shower stall when occupied. Therefore, the occupant should not also be included elsewhere in the G-189A system model during the time the shower is occupied.

f. When the water is turned on, the total (primary and secondary) water inlet and outlet flowrates are equal. When the water is turned off with air flow only, the outlet water vapor flow is equal to the inlet flow plus the amount of water evaporated. To preserve the overall water balance, the moisture retained in the towel and lost by door opening should be subtracted from the total water usage in GPOLY logic as required.

g. To satisfy the total cabin heat balance, the heat transferred through the shower stall frame, R(53), should be added to the associated cabin sensible heat addition [cabin component location R(66)] in GPOLY logic each iteration.

3.4.2.2 Heat Loss V-Array Data

Reference Location	Description	Data Type
51	Effective average shower stall frame temperature (°F)	I(0), 0
52	Effective summed conductance between shower stall and ambient surroundings (Btu/hr-°F)	0
53	Shower stall frame heat loss to ambient, excluding heat transfer to air and water flowing through stall (Btu/hr)	0
54	Ambient gas temperature (°F)	I(0)
55	Thermal convection conductor between shower stall external surface and ambient surroundings (Btu/hr-°F)	I(0)

3.4.2.2 (Continued)

Reference Location	Description	Data Type
56	Convective heat loss from shower stall outer surface to ambient gas (Btu/hr)	0
57	Ambient wall temperature (°F)	I(0)
58	Thermal radiation conductor (A3) between ambient walls and shower stall (sq ft)	
59	Radiative heat loss from shower stall to ambient walls (Btu/hr)	0
60	Temperature of ambient structure connected to shower stall (°F)	I(0)
61	Thermal conductor from shower stall to attached ambient structure (Btu/hr °F)	I(0)
62	Thermal loss from shower stall to attached ambient structure (Btu/hr)	0
63	Same as R(51) on previous iteration	0
64	Thermal conductor through insulation around shower stall. No insulation is assumed, and this value must be set to zero.	I(R)

3.4.2.3 Steady-State K-Array Data

Reference Location	Description	Data Type
16	This location is used to store flow condition on previous iteration. 0 = shower in use with water on; 1 = shower in use with air flow only; -1 = shower not in use	0

3.4.2.4 Steady-State and Transient V-Array Data (locations used for transient runs only are marked with an asterisk)

Reference Location	Description	Data Type
65	Not used	
66	Crewman metabolic rate when shower is occupied (Btu/hr)	I(0)
67	CO_2 generation rate of shower occupant (lb/hr)	0
68	Respiratory quotient of shower occupant (moles CO_2 produced/moles O_2 used)	I(0)
69	Shower stall net internal void volume when occupied (cu ft)	I(0)
70	Oxygen usage rate of shower occupant (lb/hr)	0
71	Thermal radiation conductor A3 when shower is occupied, shower wall-to-occupant (sq ft)	I(0)
72	CO_2 partial pressure in shower stall (mm Hg)	0
73	Oxygen partial pressure in shower stall (psia)	0
74	Effective thermal conductor (conduction and convection) between shower occupant and walls when occupied (Btu/hr °F)	I(0)
75	Water evaporation correction factor: ratio of actual evaporation rate to predicted value from equation (3.4.18); (dimensionless)	I(0)
76	Total electrical power (for lights) in shower stall when in use (watts)	I(0)

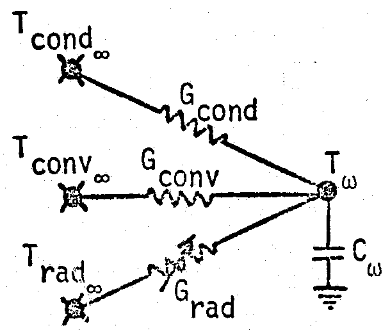
3.4.2.4 (Continued)

Reference Location	Description	Data Type
77	Shower occupant actual skin temperature (°F)	0
78	Effective surface area of shower occupant skin for air drying (sq ft)	I(0)
79	Effective heat transfer coefficient within shower, air-to-occupant/walls (Btu/hr sq ft °F)	I(0)
80	Diffusion coefficient for water vapor in air at 77°F and 14.7 psia (sq ft/hr)	I(0)
81	Shower stall total inside wall surface area for evaluating thermal convection (sq ft)	I(0)
82	Fraction of shower stall inside wall area that is wetted (for evaluating evaporative drying)	I(0)
83	Water evaporation rate inside shower stall during drying (lb/hr)	0
*84	Shower stall frame thermal capacitance (Btu/°F)	I(0)
*85	Mass of shower occupant when occupied (lb)	I(0)
*86	Maximum transient computing time increment (seconds)	I(0)
*87	Shower occupant initial skin temp (°F)	I(0)
88	This location is used to store the outlet water vapor partial pressure on the previous iteration (psia)	0

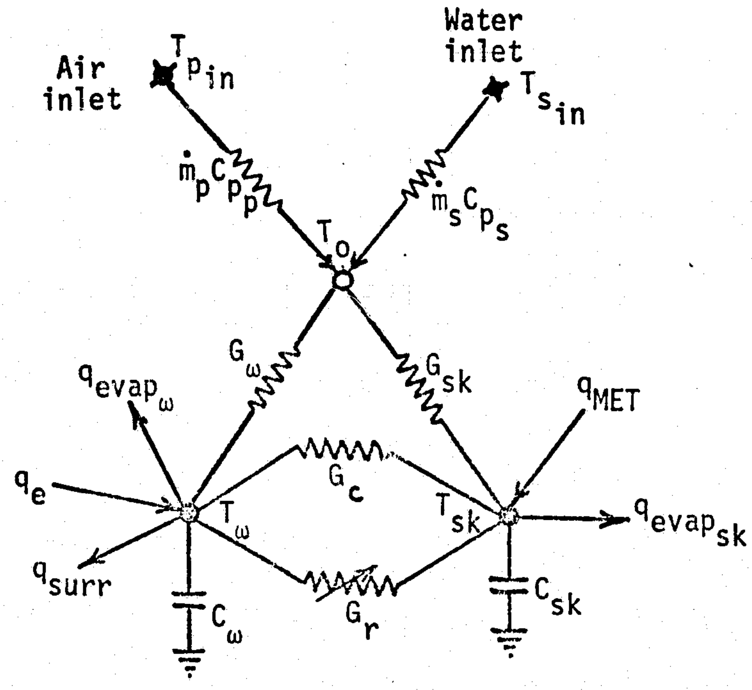
3.4.3 Analytical Model Description

3.4.3.1 Thermal Balance

The thermal nodal network assumed for the shower component is shown in Figure 3-10 for the occupied and unoccupied cases. The shower is assumed to be occupied if and only if there is flow inlet to the primary (air) side.



(a) Unoccupied



(b) Occupied

Figure 3-10. Thermal Model of Shower Stall Component

3.4.3.1.1 Unoccupied Case

Considering the unoccupied case first, an effective ambient surrounding temperature T_∞ is first found. To do this, the radiation conductor is first linearized as follows:

$$G_{rad} \equiv \sigma A \mathcal{F} \left(T_\omega^2 + T_{rad_\infty}^2 \right) \left(T_\omega + T_{rad_\infty} \right) \tag{3.4.1}$$

3.4.3.1.1 (Continued)

where temperatures are all in absolute degrees Rankine. The effective ambient temperature is then given by:

$$T_\infty = \frac{G_{rad}\, T_{rad_\infty} + G_{conv}\, T_{conv_\infty} + G_{cond}\, T_{cond_\infty}}{G_{rad} + G_{conv} + G_{cond}} \tag{3.4.2}$$

For the steady-state case, the shower wall temperature will be equal to the effective ambient temperature T_∞.

For a transient solution, the effective thermal conductance to the ambient temperature T_∞ is computed as

$$G_{eff} = G_{rad} + G_{cond} + G_{conv} \tag{3.4.3}$$

Note that equation (3.4.3) requires that all three thermal conductors be connected directly to the shower wall node. This is a slight deviation from the standard usage of the G-189A subroutine QSURR, and requires that the insulation conductance [R-array location R(64)] be set to zero. This restriction presents no problem since no insulation is planned for use on the zero-gravity whole body shower currently designed for space use. The differential equation for heat transfer from the unoccupied shower stall may then be expressed as:

$$dQ = C_\omega\, dT_\omega = -\, G_{eff}\, (T_\omega - T_\infty)\, dt \tag{3.4.4}$$

$$\frac{dT_\omega}{dt} + \left(\frac{G_{eff}}{C_\omega}\right) T_\omega = \frac{G_{eff}\, T_\infty}{C_\omega} \tag{3.4.5}$$

The solution to this equation is:

$$T_{\omega_{t=t_o+\Delta t}} = T_\infty + \left(T_{\omega_{t=t_o}} - T_\infty\right) e^{-\left(\frac{G_{eff}}{C_\omega}\right)\Delta t} \tag{3.4.6}$$

3.4.3.1.1 (Continued)

For the transient case, the unoccupied shower stall temperature is computed from equations (3.4.1), (3.4.2), and (3.4.6).

3.4.3.1.2 Occupied Case

When the shower is occupied (inlet air flow is present), the thermal model in Figure 3-10(b) applies. The basic thermal balance for the shower wall and occupant skin temperatures is the same whether the water is on or off. New temperatures are computed as follows:

$$T_{new} = T_{old} + \gamma \sum q_{into\ node} \tag{3.4.7}$$

$$\sum q_{\omega} = q_e - q_{evap_{\omega}} - q_{surr} + q_{sen_{\omega}} + G_c(T_{sk} - T_{\omega}) + \sigma G_r(T_{sk}^{\ 4} - T_{\omega}^{\ 4}) \tag{3.4.8}$$

$$\sum q_{sk} = q_{MET} - q_{evap_{sk}} + q_{sen_{sk}} + G_c(T_{\omega} - T_{sk}) + \sigma G_r(T_{\omega}^{\ 4} - T_{sk}^{\ 4}) \tag{3.4.9}$$

For steady-state cases, the value used for γ in equation (3.4.7) is

$$\gamma_{steady\ state} = \frac{0.4}{\text{sum of conductors to ground}} \tag{3.4.10}$$

Internal steady-state iterations are made until the nodal temperature change computed is 0.005°F or until 25 iterations have been performed.

For a transient solution, the value of γ is given by

$$\gamma_{transient} = \frac{\Delta t}{m\ C_p} \tag{3.4.11}$$

where $m\ C_p$ is the nodal thermal mass. The computing time increment Δt is taken to be 0.4 times the stability time increment where

$$\Delta t_{stability} = \frac{\text{thermal mass}}{\text{sum of conductors to ground}} \tag{3.4.12}$$

3.4.3.1.2 (Continued)

Initial runs with a 90-second computing time step had computational instability problems. This was overcome by further limiting the internal computing time increment to a user input value in R(86). A default value of 10 seconds is set by the subroutine for R(86) and was found satisfactory for the shower cases considered. If the system time step is less than the stability time increment, the system time step is used directly for Δt. If it is greater, then internal iterations of equations (3.4.7) through (3.4.9) are repeated until the system time step is reached.

Water turned on

When the water is turned on, it is assumed that the outlet air/water mixture is saturated. The fluid is assumed to have intimate contact with both the shower stall walls and occupant equally. Thus, its outlet temperature T_o will be the arithmetic average temperature of walls T_ω and skin T_{sk}. Therefore, for this case the thermal resistance represented by conductors G_ω and G_{sk} in Figure 3-10(b) is neglected. The sensible heat input from the fluid to the shower wall and occupant in equations (3.4.8) and (3.4.9) is thus given by

$$q_{sen_\omega} = \tfrac{1}{2}\dot{m}_p c_{p_p} (T_{p_{in}} - T_\omega) + \tfrac{1}{2}\dot{m}_s c_{p_s} (T_{s_{in}} - T_\omega) \qquad (3.4.13)$$

$$q_{sen_{sk}} = \tfrac{1}{2}\dot{m}_p c_{p_p} (T_{p_{in}} - T_{sk}) + \tfrac{1}{2}\dot{m}_s c_{p_s} (T_{s_{in}} - T_{sk}) \qquad (3.4.14)$$

The mixing of the air and water mixture is done using the standard G-189A subroutine HBALNC. This subroutine evaluates the amount of water evaporated to give saturation conditions. The latent cooling effect from this amount of evaporation is placed 70% on the occupant skin and 30% on the walls. This 70/30 split was determined from the relative temperature and surface area of the occupant and walls.

3.4.3.1.2 (Continued)

Water turned off

When air flow is present with the water (secondary flow) off, evaporative drying is assumed to occur. The latent heat of evaporation applied to the walls and occupant skin is evaluated in Paragraph 3.4.3.2. With the convection heat transfer coefficient input in R(79), the sensible heat input from the fluid to the shower wall and occupant in equations (3.4.8) and (3.4.9) is given by

$$q_{sen_\omega} = h_c A_\omega (T_o - T_\omega) \qquad (3.4.15)$$

$$q_{sen_{sk}} = h_c A_{sk} (T_o - T_{sk}) \qquad (3.4.16)$$

where the wetted wall and skin areas are input. The outlet air temperature is computed as follows:

$$T_o = \frac{\dot{m}_p \, C_{p_p} \, T_{p_{in}} + h_c \, A_\omega \, T_\omega + h_c \, A_{sk} \, T_{sk}}{\dot{m}_p \, C_{p_p} + h_c \, (A_\omega + A_{sk})} \qquad (3.4.17)$$

3.4.3.2 Evaporative Drying

When air flow is on and the water is turned off, the evaporative mass transfer coefficient is evaluated according to the following relation:

$$h_D = 0.43 c_M \left(\frac{h_c}{C_{p_a} \, \rho_a} \right) \left(\frac{Pr}{Sc} \right)^{0.67} \qquad (3.4.18)$$

$$Pr \equiv \frac{C_{p_a} \, \mu_a}{k_a} \qquad Sc \equiv \frac{\mu_f}{\rho_a \, D} \qquad (3.4.19)$$

3.4.3.2 (Continued)

The water vapor diffusion coefficient D is input for 14.7 psia and 77°F. To adjust for the actual conditions, the following equation from Reference 5 is used:

$$D \propto \frac{T^{1.75}}{P} \tag{3.4.20}$$

where T and P are in absolute units. The constant 0.43 in equation (3.4.18) was determined by correlating actual shower runs with experimental data. If further correlation is required, the multiplying factor c_M was added, which is input in R(75). If c_M is not input, the subroutine assumes a default value of 1.0. The evaporation rate is then given by

$$\dot{m}_{evap} = \frac{h_D A}{R_{H20}} \left[\frac{P_{H20_{surface}}}{(T_{surface} + 460)} - \frac{P_{H20_o}}{(T_o + 460.)} \right] \tag{3.4.21}$$

and the latent cooling is given as

$$q_{evap} = 1042\ \dot{m}_{evap} \tag{3.4.22}$$

3.4.3.3 Crewman Metabolic Simulation

The thermal, metabolic oxygen, and CO_2 effects of the crewman are accounted for within the shower stall when occupied. Therefore, the occupant should not also be included elsewhere in the G-189A model during the time the shower is occupied. The shower is considered to be occupied if and only if there is inlet flow on the primary (air) side. The occupant metabolic rate and respiratory quotient are input (default values of 650 Btu/hr and 0.9 will be used if these data are not input). As recommended by Reference 7, one third of the crewman thermal mass is used as a skin thermal node. Only the thermal effects on this node are included, as the central

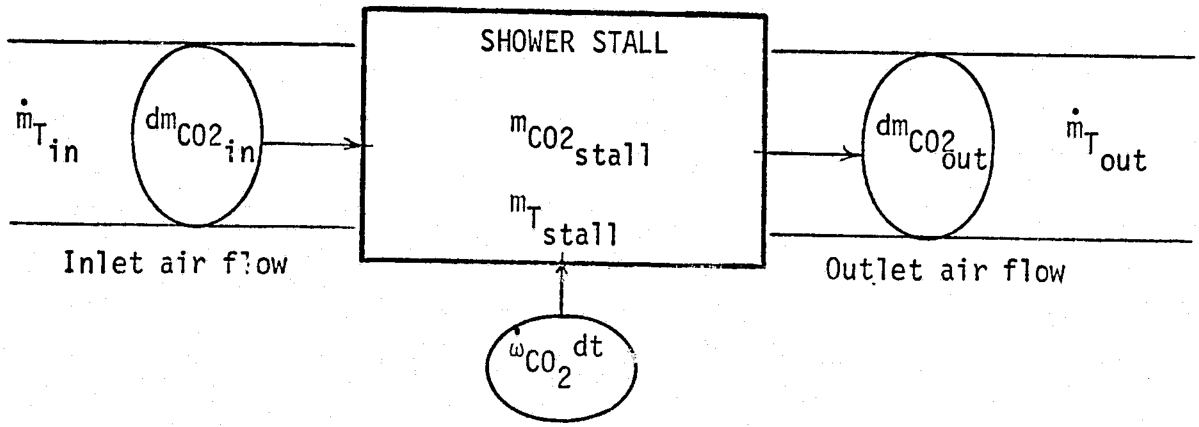
3.4.3.3 (Continued

core mass is assumed to remain at constant temperature. The oxygen consumption rate and carbon dioxide generation rate are computed from the relations from Reference 8:

$$\dot{u}_{O2} = Q_{MET} \times 10^{-3} \left[0.1708 - 0.0123 \left(\frac{RQ - 0.707}{0.293} \right) \right] \qquad (3.4.23)$$

$$\dot{\omega}_{CO2} = \left(\frac{44.01}{32} \right) RQ \; \dot{u}_{O2} \qquad (3.4.24)$$

The carbon dioxide balance within the shower stall is made assuming the flow model shown below:



Complete and immediate mixing is assumed within the shower enclosure. Therefore, the total outlet flow is identical in composition to the gas within the shower, or

$$\dot{m}_{CO2_{out}} = \left(\frac{m_{CO2_{stall}}}{m_{T_{stall}}} \right) \dot{m}_{T_{out}} \qquad (3.2.25)$$

A mass balance within the shower stall over a differential time increment dt is

$$\dot{m}_{CO2_{in}} + \dot{\omega}_{CO_2} - \dot{m}_{CO2_{out}} = \dot{m}_{CO2_{stall}} \qquad (3.4.26)$$

3.4.3.3 (Continued)

Combining equations (3.4.25) and (3.4.26) and rearranging;

$$\frac{d\dot{m}_{CO2_{stall}}}{dt} + \left(\frac{\dot{m}_{T_{out}}}{m_{T_{stall}}}\right) m_{CO2_{stall}} = \dot{m}_{CO2_{in}} + \dot{\omega}_{CO2} \tag{3.4.27}$$

For continuous operating conditions, it is assumed the CO_2 inlet flow rate and generation rate (and the total outlet flow) remain constant over a time step. Since the amount of CO_2 in the stall is relatively small compared to the total fluid mass, the fractional variation in total mass is assumed negligible compared with the variation of CO_2. Therefore, the solution to equation (3.4.27) is as follows:

$$\left.\begin{aligned} m_{CO2_{stall}}(t=t_o+\Delta t) &= \beta_{CO2} + \left[m_{CO2_{stall}}(t=t_o) - \beta_{CO2}\right] e^{-\left(\frac{\dot{m}_{T_{out}}}{m_{T_{stall}}}\right)\Delta t} \\ \beta_{CO2} &\equiv m_{T_{stall}} \left(\frac{\dot{m}_{CO2_{in}} + \dot{\omega}_{CO_2}}{\dot{m}_{T_{out}}}\right) \end{aligned}\right\} \tag{3.4.28}$$

For a transient solution, equation (3.4.28) is used to compute the total mass of CO_2 in the shower stall at the end of the system time step Δt. For the steady-state case, the solution at $\Delta t=\infty$ applies; whereby, $m_{CO2_{stall}}$ is equal to β_{CO2}. Knowing $m_{CO2_{stall}}$, the outlet CO_2 flow rate is obtained from equation (3.4.25). The CO_2 partial pressure within the stall is found in either case assuming a perfect gas; whereby,

$$P_{CO2} = \frac{m_{CO2_{stall}} R_{CO_2} (T + 460)}{2.785\ V} \tag{3.4.29}$$

The preceding analysis for CO_2 levels may also be made for the oxygen balance. The final result, giving the total mass of oxygen in the shower

3.4.3.3 (Continued)

after a time step Δt, is given by

$$\left.\begin{aligned} m_{O2_{stall}}(t=t_o+\Delta t) &= \beta_{O2} + \left[m_{O2_{stall}}(t=t_o) - \beta_{O2}\right] e^{-\left(\frac{\dot{m}_{T_{out}}}{m_{T_{stall}}}\right)\Delta t} \\ \beta_{O2} &\equiv m_{T_{stall}} \left(\frac{\dot{m}_{O2_{in}} - \dot{u}_{O2}}{\dot{m}_{T_{out}}}\right) \end{aligned}\right\} \quad (3.4.30)$$

Again, for the steady-state case, the mass of oxygen in the shower stall is simply β_{O2}. The outlet oxygen flow rate in either case is given by

$$\dot{m}_{O2_{out}} = \left(\frac{m_{O2_{stall}}}{m_{T_{stall}}}\right) \dot{m}_{T_{out}} \quad (3.4.31)$$

and the oxygen partial pressure within the stall is given by

$$P_{O2} = \frac{m_{O2_{stall}} \; R_{O2} \; (T + 460)}{144 \; V} \quad (3.4.32)$$

During transient operation, the input control variable K(16) is used to determine when the shower is being first turned on after being unoccupied. If it is, the oxygen and CO_2 partial pressures within the stall are reinitialized at their values for the inlet air.

3.4.3.4 Pressure Drop Considerations

The shower may be operated either with or without inlet water flow. Average shower use time is reported to be 10 minutes, with an average of 2.5 liters (0.66 gal) of water used. For a water supply flow rate of 75 kg/hr (165 lb/hr), the water will be turned on for a cumulative total of 2 minutes per shower. This will occur during several water applications. A vacuum pick-up device is located in the bottom of the shower stall to collect the water and air flow. Clearly, two-phase flow will sometimes be occurring, with the pressure drop across the vacuum collection device fluctuating widely as slugs of water are picked up with air flow. The G-189A program should not be used to predict such fluctuations due to possible computational instability problems. Therefore, when water flow is being inlet to the shower, the pressure drop of shower components should not be made.

The shower subroutine does not account for the time lag of liquid water flowing through the stall. Thus, when the inlet water is turned off, air flow only is assumed to occur, and pressure drop may be computed. The pressure drop predictions of Reference 19 were used to determine data for the shower components. The values given for the various shower components are shown in Table 3-13. To derive G-189A pressure drop model data from these figures, the method of Reference 20 was used with the following relation:

$$\frac{\rho \Delta P}{\mu^{\alpha}} \propto \dot{m}^{2-\alpha}$$

From the data presented in Reference 20, and assuming turbulent flow, a value of 0.2 for α was assumed. Thus, neglecting the air viscosity difference

3.4.3.4 (Continued)

between actual conditions and those estimated in Table 3-13, the shower component pressures may be predicted in G-189A runs from the following equation:

$$\left.\begin{aligned} \Delta P &= C' \frac{\dot{m}^{1.8}}{\rho} \\ C' &\equiv \frac{\Delta P_t \, \rho_t}{\dot{m}_t^{1.8}} \end{aligned}\right\} \quad (3.4.33)$$

where subscript t refers to the data in Table 3-13 (or any future test data which should become available), and the unsubscripted variables refer to the changing shower conditions computed during a G-189A run. Equation (3.4.33) represents G-189A standard pressure drop option (3) (see Table 3-1), and the value of C' and the exponent 1.8 must be input. Assuming the air density to be 0.075 lb/cu ft, the values of C' in equation (3.4.33) for the shower components described in Table 3-13 are as follows:

Shower Component	Value of C for G-189A Pressure Drop Option 3
Shower stall (including vacuum pick-up device, line, fittings)	4.74×10^{-5}
Air heater (with lines, flowmeter and fittings)	3.79×10^{-5}
Water/air separator-air side (with attached fittings)	8.70×10^{-5}

These values for C', together with the exponent of 1.8 in equation (3.4.33), may be input as component pressure drop model data using the G-189A option (3) in Table 3-1.

TABLE 3-12

DEFAULT VALUES FOR SHOWER SUBROUTINE INPUT DATA

R-Array* Location	Default Value
51	70
54	70
55	14
57	70
58	14
60	70
61	30
66	650
68	0.9
69	39
71	13
74	4.5
75	1
76	24
77	90
78	16.575
79	0.8
80	1
81	46
82	1
84	10
85	170
86	10
87	80

*See Section 3.4.2 for definition of variables.

TABLE 3-13

ESTIMATED PRESSURE DROP (AND RISE) FOR VARIOUS ZERO-GRAVITY WHOLE BODY SHOWER COMPONENTS

SHOWER COMPONENT	PRESSURE DROP (OR RISE)	AIR FLOW
Shower stall (including vacuum pick-up device, line, fittings)	345 N/m^2 (7.20 psf)	81.6 kg/hr (180 lb/hr)
Air heater (with lines, flowmeter and fittings)	276 N/m^2 (5.76 psf)	81.6 kg/hr (180 lb/hr)
Water/air separator (with attached fittings)	786 N/m^2 (16.42 psf)	91.9 kg/hr (202.5 lb/hr)
Blower - pressure rise indicated (with attached lines, fittings)	1407 N/m^2 (29.38 psf)	91.9 kg/hr (202.5 lb/hr)

TABLE 3-14

DEFINITION OF SYMBOLS FOR SHOWER SUBROUTINE DESCRIPTION

A = Area (sq ft)

C = Thermal mass (Btu/°F)

C_M = Multiplying factor to adjust analytical mass transfer coefficient (dimensionless)

C_p = Thermal specific heat (Btu/lb°F)

D = Diffusion coefficient for water vapor in air (sq ft/hr)

e = Naperian base logarithm, 2.7183...

$\mathfrak{F}$ = Thermal radiation gray-body shape factor (dimensionless)

G = Thermal conductor (Btu/hr-°F)

G_{cond} = R(61) = Effective thermal conductor between shower stall frame and attached ambient structure (Btu/hr °F)

G_{conv} = R(55) = Effective thermal conductor between shower stall frame and ambient gas (Btu/hr °F)

G_{eff} = Effective thermal conductor between shower stall frame and ambient surroundings (Btu/hr °F)

G_{rad} = Effective thermal radiation conductor between shower stall and ambient wall (Btu/hr °F)

h_c = Thermal convection heat transfer coefficient in shower (Btu/hr-sq ft-°F)

h_D = Evaporative mass transfer coefficient (ft/hr)

k_a = Fluid thermal conductivity in shower (Btu/hr-ft-°F)

m = mass (lbs)

$\dot{m}$ = Mass flow rate (lb/hr)

P = Absolute pressure (lb/sq ft)

P_{CO2} = R(72) = CO_2 partial pressure in shower stall (mm Hg)

TABLE 3-14 (continued)

DEFINITION OF SYMBOLS FOR SHOWER SUBROUTINE DESCRIPTION

P_{O2}	= R(73) = Oxygen partial pressure in shower stall (psia)
Pr	= Air Prandtl number in shower (dimensionless)
q_e	= 3.413 x R(76) = Total electrical power (for lights) in shower stall (Btu/hr)
q_{evap}	= Latent heat of evaporation (Btu/hr)
q_{MET}	= R(66) = Shower occupant metabolic rate (Btu/hr)
q_{surr}	= R(53) = Shower stall frame heat loss to ambient, excluding heat transfer to air and water flowing through stall (Btu/hr)
R	= Ideal gas constant (ft-lb/lb °R)
RQ	= R(68) = Metabolic respiratory quotient of shower occupant (moles CO_2 produced/moles O_2 used)
Sc	= Schmidt number for shower air (dimensionless)
T	= Temperature (°F, except where otherwise noted)
t	= Mission time (hours)
T_{cond_∞}	= R(60) = Temperature of ambient structure connected to shower frame (°F)
T_{conv_∞}	= R(54) = Ambient gas temperature surrounding shower enclosure (°F)
t_o	= Mission time at beginning of current time step (hours)
T_{rad_∞}	= R(57) = Effective average ambient wall temperature surrounding shower enclosure (°F unless otherwise specified)
$\dot{u}_{O2}$	= Metabolic oxygen consumption rate of shower occupant (lb/hr)
V	= Shower stall internal void volume (cu ft)
β	= Constant in metabolic balance equation (3.4.28 , 30)
σ	= Stefan-Boltzmann constant (0.1714×10^{-8} Btu/hr-sq ft- $°R^4$)
Δt	= Transient solution time increment (hours)

TABLE 3-14 (concluded)

DEFINITION OF SYMBOLS FOR SHOWER SUBROUTINE DESCRIPTION

μ = Air viscosity in shower (lb/hr-ft)

ρ = Air density in shower (lb/cu ft)

γ = Multiplier to compute new temperatures, defined by equation (3.4.7)

$\dot{\omega}_{CO2}$ = Metabolic carbon dioxide generation rate of shower occupant (lb/hr)

Subscripts

a = Shower air
c,cond = Thermal conduction
conv = Thermal convection
CO2 = Carbon dioxide
evap = Evaporation
H2O = Water vapor
in = Inlet condition
new = Condition at end of current time step or iteration
o = Shower air outlet condition
old = Condition at beginning of current time step or iteration
O2 = Oxygen
p = Primary side
r,rad = Thermal radiation
s = Secondary side
sen = Sensible heat
sk = Shower occupant skin node
stall = Condition inside shower stall
T = Total fluid property
∞ = Ambient condition surrounding shower enclosure
ω = Shower stall walls/frame node

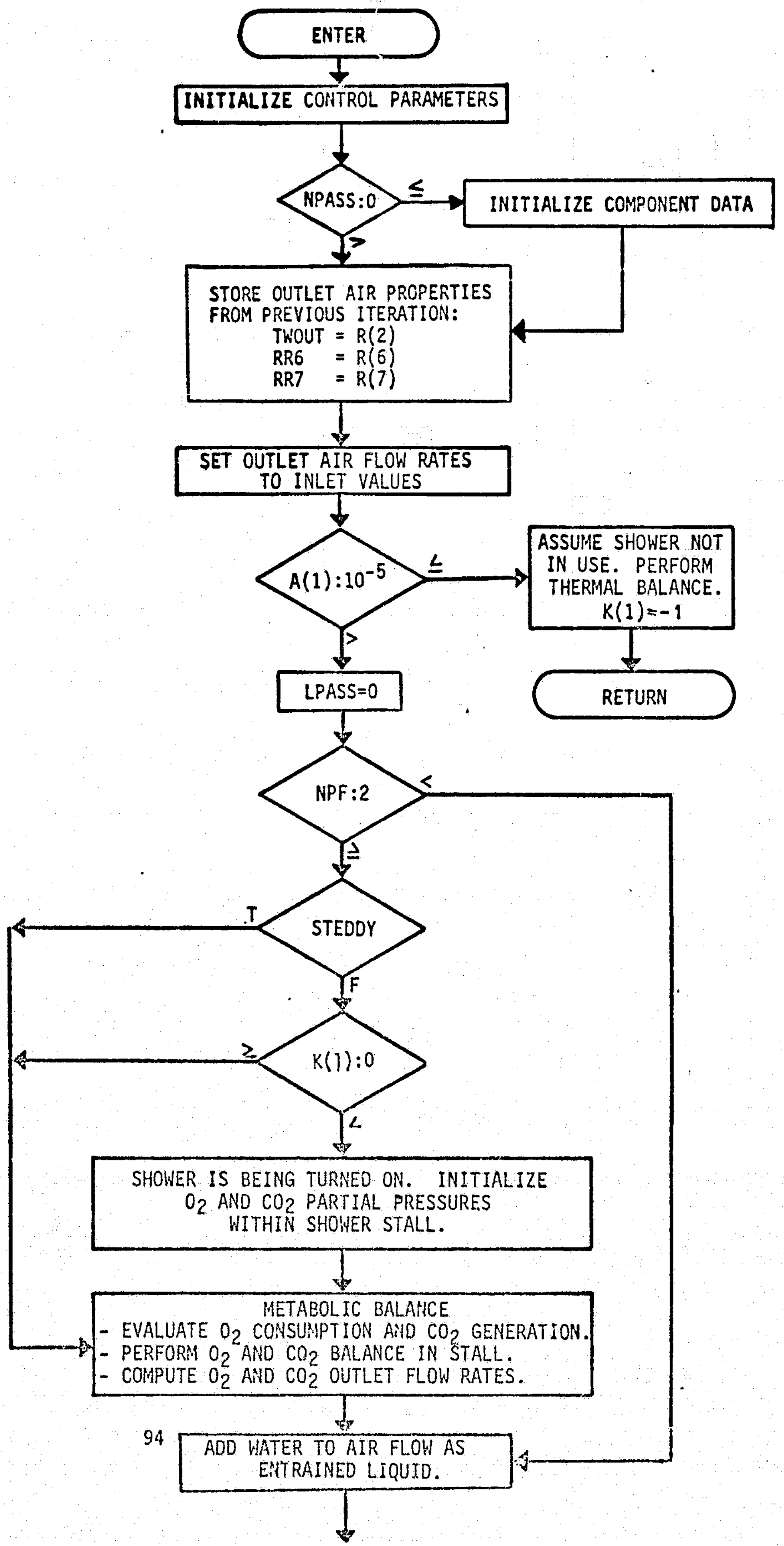


Figure 3-11. Logic Flow Chart for SHOWER Subroutine

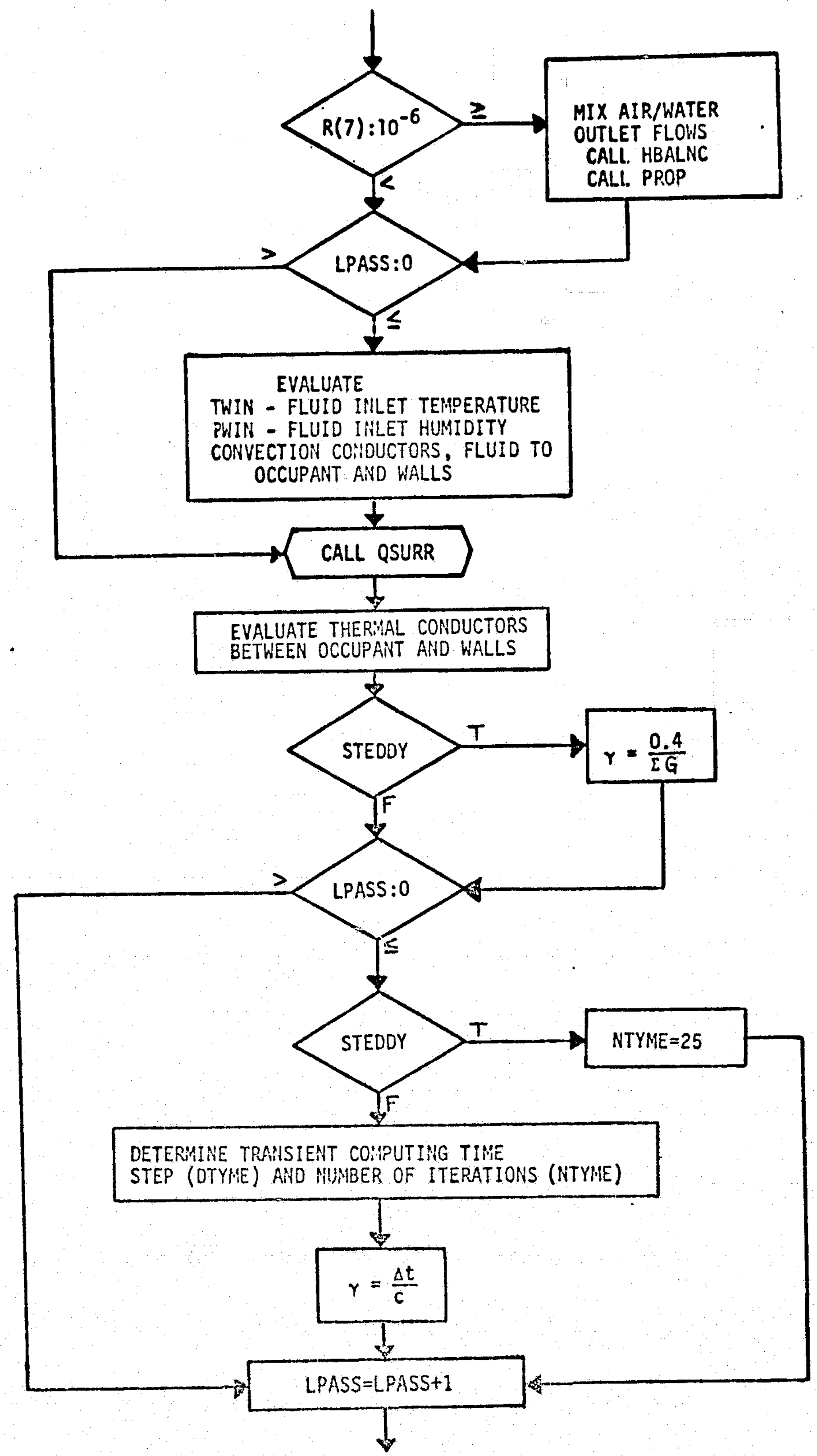


Figure 3-11. Logic Flow Chart for SHOWER Subroutine (continued)

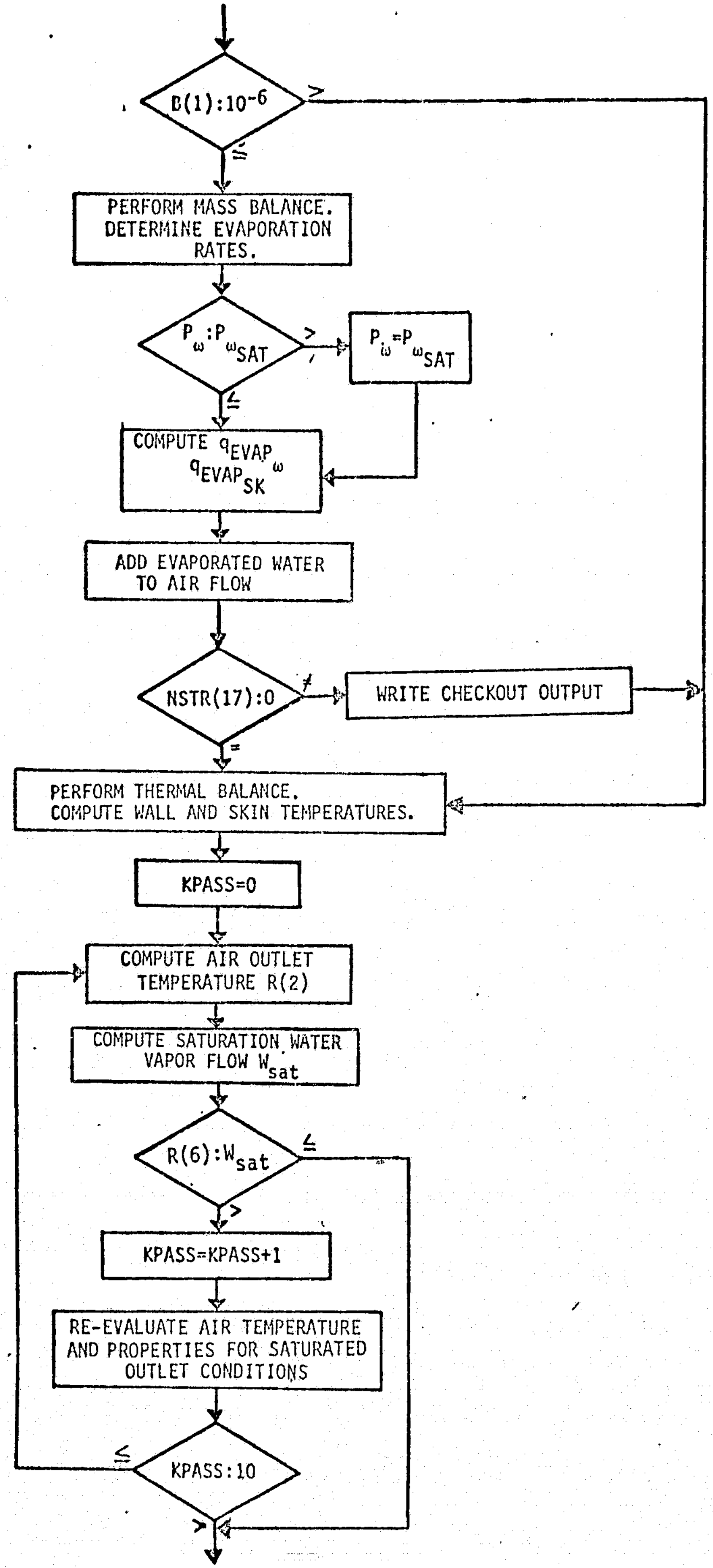


Figure 3-11. Logic Flow Chart for SHOWER Subroutine (continued)

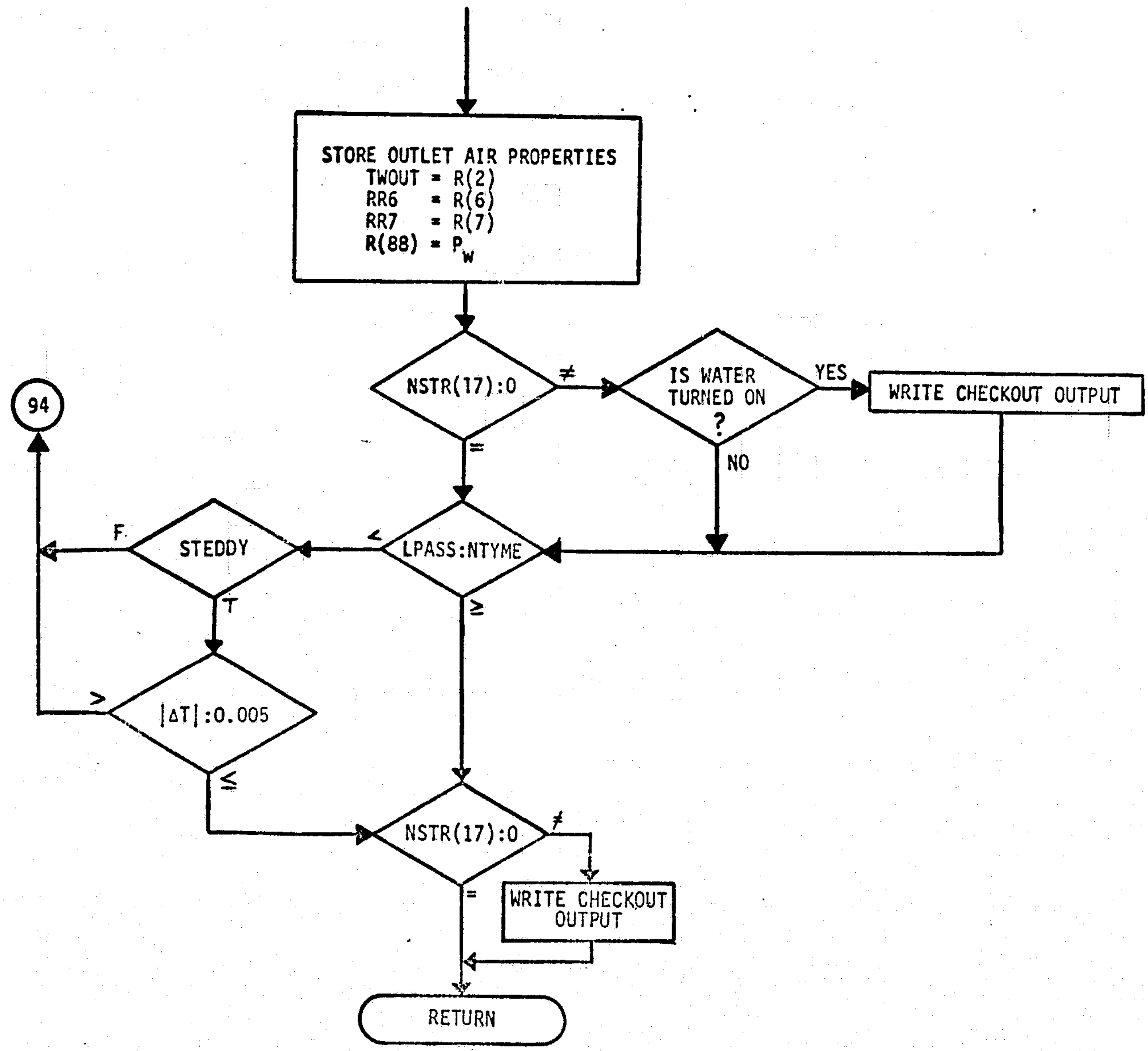


Figure 3-11. Logic Flow Chart for SHOWER Subroutine (concluded)

TABLE 3-15

DEFINITION OF SHOWER SUBROUTINE VARIABLES

CPAA	=	Variable used to store inlet air specific heat (Btu/lb °F)
CSKIN	=	Effective thermal capacitance of shower occupant skin node (Btu/hr °F)
CWALL	=	Effective thermal capacitance of shower wall node (Btu/hr °F)
DIFF	=	Water vapor diffusion coefficient in air inside shower stall (sq ft/hr)
DTH	=	System transient time step (hours)
DTYME	=	Internal transient computing time step (hours)
EVAPSK	=	Water evaporation rate on shower occupant (lb/hr)
EVAPW	=	Water evaporation rate on shower walls (lb/hr)
FIRST	=	Logical variable set to TRUE for first internal iteration with water either on or off; otherwise FALSE
GRAD	=	Linearized radiation thermal conductor from shower stall walls to ambient wall (when shower is unoccupied) or to occupant (if shower is occupied); (Btu/hr-°F)
GSK	=	Sum of thermal conductors to shower occupant (Btu/hr °F)
GSKIN	=	Thermal conductor from inlet air/water to shower occupant (Btu/hr °F)
GW	=	Sum of thermal conductors to shower stall (Btu/hr °F)
GWALL	=	Thermal conductor from inlet air/water to shower walls (Btu/hr °F)
H51,H77	=	Net heat input from fluid to wall and skin nodes respectively (Btu/hr)
HC	=	Heat transfer coefficient; air to shower occupant and walls (Btu/hr-sq ft-°F)
HD	=	Mass transfer coefficient for water evaporation inside shower (ft/hr)
HMIX	=	Total enthalpy of outlet air/water mixture (Btu/hr)
KPASS	=	Counter for number of iterations to determine saturated conditions
LPASS	=	Internal heat and mass balance iteration counter

TABLE 3-15 (concluded)

DEFINITION OF SHOWER SUBROUTINE VARIABLES

NTYME = Maximum number of internal heat and mass balance iterations required

PR = Air Prandtl number inside shower (dimensionless)

PVAP = Water vapor partial pressure resulting from mass transfer equation (psia)

PWIN = Water vapor partial pressure of inlet air (psia)

PWINF = Effective water vapor pressure within shower stall (psia)

PWOUT = Outlet water vapor pressure computed on previous iteration (psia)

QSKIN = Latent heat of evaporation on shower occupant (Btu/hr)

QWALL = Latent heat of evaporation on shower walls (Btu/hr)

R1,R2,R6,R10,R12 = Variables used to store inlet air total flow, temperature, humidity, O_2 and CO_2 flows during current iteration

R51,R77 = Variables used to store wall and skin node temperatures during heat balance

RHOAA = Variable used to store inlet air density (lb/cu ft)

RR6,RR7 = Variables used to store component outlet water vapor and liquid flows from previous system iteration

SC = Air Schmidt number inside shower (dimensionless)

TWIN = Inlet air temperature (°F)

TWINF = Effective air temperature within shower stall (°F)

TWOUT = Air/water outlet temperature (°F)

VISCAA = Variable used to store inlet air viscosity (lb/hr ft)

WCO2 = Weight of CO_2 in shower stall (lb)

WOXY = Weight of oxygen in shower stall (lb)

X51,X77 = Heat balance multiplier for computing new wall and skin node temperatures, respectively, defined by equation (3.4.7)

XKAA = Variable used to store inlet air thermal conductivity (Btu/hr ft °F)

3.5 WASDRY

The WASDRY subroutine is designed as G-189A No. 70.

3.5.1 Subroutine Description

This subroutine will simulate operation of the following appliances:

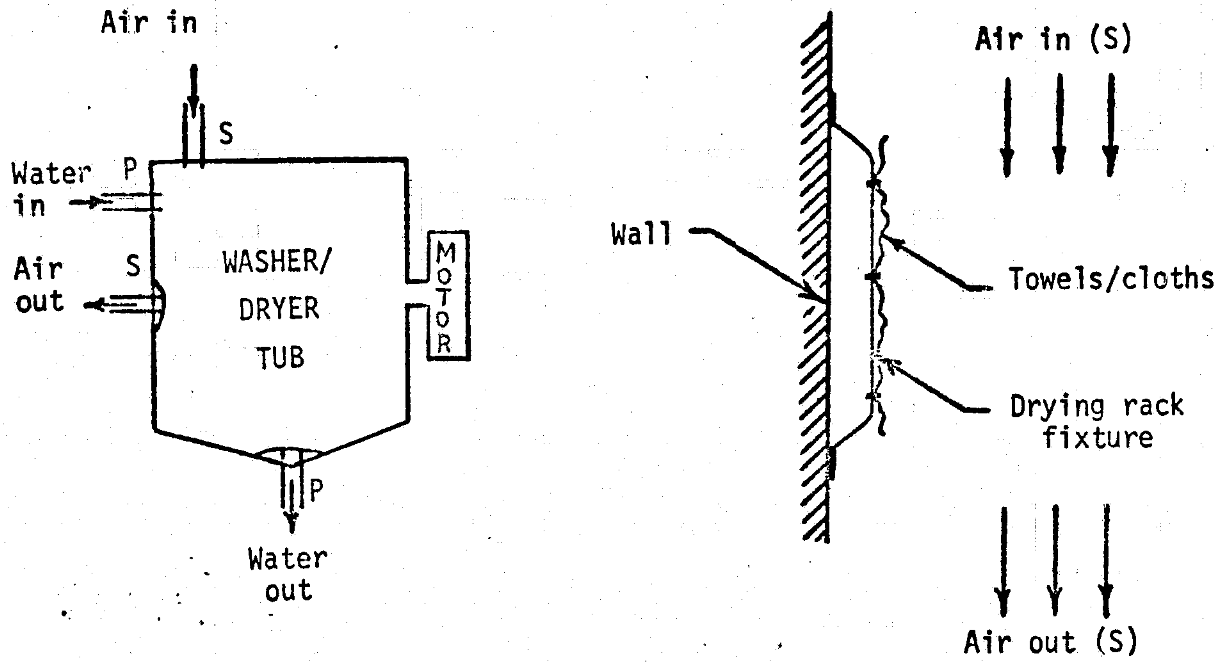
- o Clothes washer
- o Clothes dryer
- o Clothes washer/dryer combination
- o Dishwasher
- o Dish dryer
- o Dishwasher/dryer combination
- o Towel/cloth drying rack

A flow schematic of the WASDRY component is shown in Figure 3-12. Seven operational usage phases may be simulated:

Phase 0 - Unit off
Phase 1 - Wash water fill
Phase 2 - Wash (circulate)
Phase 3 - Spin dry - wash water out
Phase 4 - Rinse water fill
Phase 5 - Rinse (circulate)
Phase 6 - Spin dry - rinse water out
Phase 7 - Dry

The routine will control the switching between phases based on input cycle schedules if requested by the user. Thermal exchange between the tub and frame and the ambient environment is modeled using the standard G-189A library routine QSURR. During the drying phase, the evaporation process is modeled in detail as a function of air inlet flow rate, humidity and properties, velocity within the tub, and water retention in the load. The subroutine models the thermal/mass exchange in the agitator or tub only, with the peripheral pumps, valves, accumulator, etc. to be simulated by standard G-189A component routines.

3.5.1 (Continued)



(a) Washer/dryer

(b) Towel/cloth rack dryer

Figure 3-12. WASDRY Component Flow Schematic

3.5.2 Subroutine Data

3.5.2.1 General Notes

a. Water for the wash cycle is inlet and outlet on the component primary side. All liquid flow codes (0 and 4) may be used. Air for drying is inlet and outlet on the component secondary side. All gaseous flow codes (1, 2, or 3) may be used.

b. If a towel/cloth rack dryer is being simulated, only phase 0 and 7 may be used.

c. To satisfy the total cabin heat balance, the heat lost through the washer structure, R(53), should be added to the associated cabin

3.5.2.1 (Continued)

component sensible heat addition [cabin component location R(66)] in GPOLY logic each iteration.

3.5.2.2 Instruction Options

NSTR(1): Type of unit to be simulated

- = 0 Washer and dryer combination
- = 1 Washer only
- = 2 Dryer only
- = 3 Towel/washcloth drying rack

NSTR(2): Steady-state criterion to be used for dryer

- = 0 Constant evaporation rate assumed equal to total water evaporated [derived from R(77), R(94) and R(78)] divided by total time allowed in R(79) for drying. Note: This value is limited internally to a drying rate corresponding to saturated outlet conditions.
- = 1 Assume saturated dryer outlet conditions
- = 2 Use normal transient dryer evaporation equations, while holding the residual water loading constant

NSTR(3): Maximum air absolute velocity in dryer tub

- = 0 Use input value in R(80)
- = 1 Use air inlet velocity based on flow rate and cross-sectional area in R(81)
- = 2 Compute velocity based on tub outer diameter and rotational speed in R(82) and R(83)
- = 3 Same as 2 above, except limited to value in 1 above

3.5.2.2 (Continued)

NSTR(4): Type of load

= 0 Clothes

= 1 Dishes

NSTR(5): Cycle phase logic request for transient run

= 0 Do not perform cycle phase logic. User will control the cycle phase in R(95)

= 1 Perform cycle phase logic. Cycle phase to be used each time step will be determined and output in R(95)

3.5.2.3 Heat Loss V-Array Data

Reference Location	Description	Data Type
51	Effective average unit frame temperature [or drying rack fixture temperature if NSTR(1)=3]; (°F)	I(0), 0
52	Effective summed conductance from unit frame (or drying rack fixture) to ambient surroundings (Btu/hr °F)	0
53	Total heat loss from unit frame (or drying rack fixture) to ambient surroundings (excluding heat transfer to tub and clothes/dishes load); (Btu/hr)	0
54	Ambient gas temperature (°F)	I(0)
55	Thermal convection conductor between unit frame external surface (or drying rack fixture) and ambient gas (Btu/hr °F)	I(0)
56	Convective heat loss from unit frame (or drying rack fixture) to ambient gas (Btu/hr)	0
57	Ambient wall temperature (°F)	I(0)

3.5.2.3 (Continued)

Reference Location	Description	Data Type
58	Thermal radiation conductor A∋ between unit frame (or drying rack fixture) and ambient walls (sq ft)	I(0)
59	Radiative thermal loss from unit frame (or drying rack fixture) to ambient walls (Btu/hr)	0
60	Temperature of ambient structure attached to unit frame (or drying rack fixture); (°F)	I(0)
61	Conductive thermal conductor between unit frame (or drying rack fixture) and attached ambient structure (Btu/hr °F)	I(0)
62	Conductive thermal loss from unit frame (or drying rack fixture) to attached ambient structure (Btu/hr)	0
63	External surface temperature of insulation (if any) around unit frame (°F). Same as R(51) if no insulation is used.	0
64	Thermal conductor through insulation around unit frame (Btu/hr °F). May be zero if no insulation is used, and should be zero for towel/washcloth drying rack fixture.	I(R)

3.5.2.4 Steady-State and Transient K-Array Data

Reference Location	Description	Data Type
16	Steady-state operational cycle phase to be used	I(R)
17	Specifies next mission time the unit is to be turned on; may be either 1, 2, or 3, corresponding to being first turned on at mission time in R(99), R(100), or R(101), respectively.	I(0) if NSTR(5)=1, 0

3.5.2.4 (Continued)

Reference Location	Description	Data Type
18	Cycle phase used for previous iteration	0

3.5.2.5 Steady-State and Transient V-Array Data

Note: (1) Locations used for transient runs only are marked with an asterisk.

(2) All locations used for towel/cloth drying rack [NSTR(1)=3] are marked "--".

Reference Location	Description	Data Type
65--	Heat input to component from drum motor (Btu/hr)	0
66--	Clothes or dishes load, dry (or weight of dry clothes on drying rack); (lbs)	I(O)
67--	Weight of water in tub (or in clothes for drying rack); (lbs). This value used for steady-state analysis only.	I(R)
68	Drum motor input electrical power for continuous wash/rinse/dry (watts)	I(O)
69	Not used	
70	Drum motor input electrical power for high-speed spin dry (watts)	I(O)
71	Not used	
72--	Tub and contents temperature (°F); (or temperature of clothes on drying rack)	I(O), O
73	Weight of detergent used per wash (lb detergent per lb water). Note: This item may only be used if the primary flow code is 4 since the detergent in the outlet wash water is put in the Special Flow No. 3.	I(R)
74	Not used	

3.5.2.5 (Continued)

Reference Location	Description	Data Type
*75--	Effective average specific heat of clothes or dishes load-dry (Btu/lb °F)	I(0)
76--	Thermal conductor between tub and unit frame (or between towels/cloths and drying rack fixture); (Btu/hr °F)	I(0)
77--	Residual water retention after spin-dry (or initial water loading for towels/cloths on drying rack), used for clothes load only (lb water/lb dry clothes)	I(0) if NSTR(4)=0
78--	Residual water retention required to terminate the drying process (lb water/lb dry clothes or dishes)	I(0) if NSTR(1)≠1
79--	Maximum time to be allowed for drying process (minutes). Note: For automatic control [NSTR(5)=1], the drying phase may be terminated sooner, depending on the amount of water retained in the load.	I(0) if NSTR(1)≠1
80--	Effective maximum air absolute velocity in drum during drying (ft/hr); or air velocity over towel/cloth drying rack.	I(0) if NSTR(1)≠1 and NSTR(3)=0, 0
81	Total cross-sectional area of air inlet to dryer drum (sq in)	I(0) if NSTR(1)≠1 and NSTR(3)=1 or 3
82	Diameter of dryer drum (inches)	I(0) if NSTR(1)≠1 and NSTR(3)=2 or 3
83	Rotational speed of dryer drum (rpm)	I(C) if NSTR(1)≠1 and NSTR(3)=2 or 3
84	Not used	
85	Not used	

3.5.2.5 (Continued)

Reference Location	Description	Data Type
86	Ratio of effective average air velocity in dryer drum relative to clothes or dishes load to maximum absolute air velocity in R(80)(dimensionless)	I(0) if NSTR(1)≠1
87--	Effective exposed surface area of clothes or dishes load for drying (sq ft/lb dry load)	I(0) if NSTR(1)≠1
88--	Diffusion coefficient for water vapor in air at 77°F and 14.7 psia (sq ft/hr)	I(0) if NSTR(1)≠1
89--	Experimental correlation factor to adjust analytical evaporative drying rate (dimensionless)	I(0)
90	Not used	
91	This location used to store outlet water vapor partial pressure on previous iteration (psia)	0
92--	Water evaporation rate during drying before adjusting for capillary action/surface tension effects (lb/hr)	0
93--	Actual water evaporation rate during drying (lb/hr)	0
94	Residual water loading after spin-dry, used for dishes load only (lb water/lb dry dishes)	I(0) if NSTR(4)=1
*95--	Current cycle phase to be simulated during transient run	I(R). Also output if NSTR(5)=1
*96--	Weight of water in tub and clothes/dishes load during transient run (lbs); or water retained in towels/cloth for rack dryer	I(R), 0
97	Time to be allowed for wash cycle (minutes)	I(0) if NSTR(5)=1

3.5.2.5 (Continued)

Reference Location	Description	Data Type
98	Time to be allowed for rinse cycle (minutes)	I(0) if NSTR(5)=1
	Note: The following three cycle start times are automatically repeated every 24 hours. Zero start times for R(100) and R(101) are allowable and will be ignored.	
*99	First mission time (hrs) for unit to turn on	I(R) if NSTR(5)=1
*100	Second mission time (hrs) for unit to turn on	I(R) if NSTR(5)=1
*101	Third mission time (hrs) for unit to turn on	I(R) if NSTR(5)=1
102	Total quantity of water to be used during wash cycle (lb)	I(0) if NSTR(5)=1
103	Total quantity of water to be used during rinse cycle (lbs)	I(0) if NSTR(5)=1
*104--	Internal computing time increment used (hrs)	0
*105--	Maximum computing time increment allowed for computing evaporative drying (sec)	I(0)
*106--	Thermal mass of empty tub only (Btu/°F); input zero for rack dryer	I(0)
*107--	Thermal mass of unit frame (Btu/°F); or of towel/cloth drying rack fixture	I(0)
*108	This location is used to store the mission start time of the current cycle phase (sec), used for cycle phase determination logic	0

3.5.2.5 (Continued)

Reference Location	Description	Data Type
109--	Critical moisture content for water evaporation from clothing load (lb water/lb dry clothes). For clothes, this value represents the point where capillary action/surface tension effects begin to retard the evaporation from a free liquid surface. For dishes, this value represents the point where the initially complete wetted area begins to decrease.	I(0) if NSTR(1)≠
110	Time to be allowed for spin-dry (min)	I(0) if NSTR(5)=1

3.5.3 Analytical Model Description

3.5.3.1 Thermal Balance

Nodes having thermal mass

The thermal model assumed for the WASDRY component is shown in Figure 3-13. The temperatures of the washer/dryer tub and frame (or towel rack and cloths) are computed as follows:

$$T_{new} = T_{old} + \gamma \, \Sigma q_{into\ node} \tag{3.5.1}$$

For a washer/dryer component, the net heat into the tub and frame is:

$$\Sigma q_t = \dot{m}_s \, C_{p_s} \left(T_{a_{in}} - T_t \right) + \dot{m}_p C_{p_p} \left(T_{\omega_{in}} - T_t \right) - q_{evap} + G_{tf} \, (T_f - T_t)$$

$$\Sigma q_f = G_{tf} \, (T_t - T_f) + q_{MOTOR} - q_{surr}$$

For a towel/cloth drying rack, the net heat into the cloth and fixture is:

$$\Sigma q_t = G_a \left(T_{a_{out}} - T_t \right) - q_{evap} + G_{tf} \, (T_f - T_t)$$

$$\Sigma q_f = G_{tf} \, (T_t - T_f) - q_{surr}$$

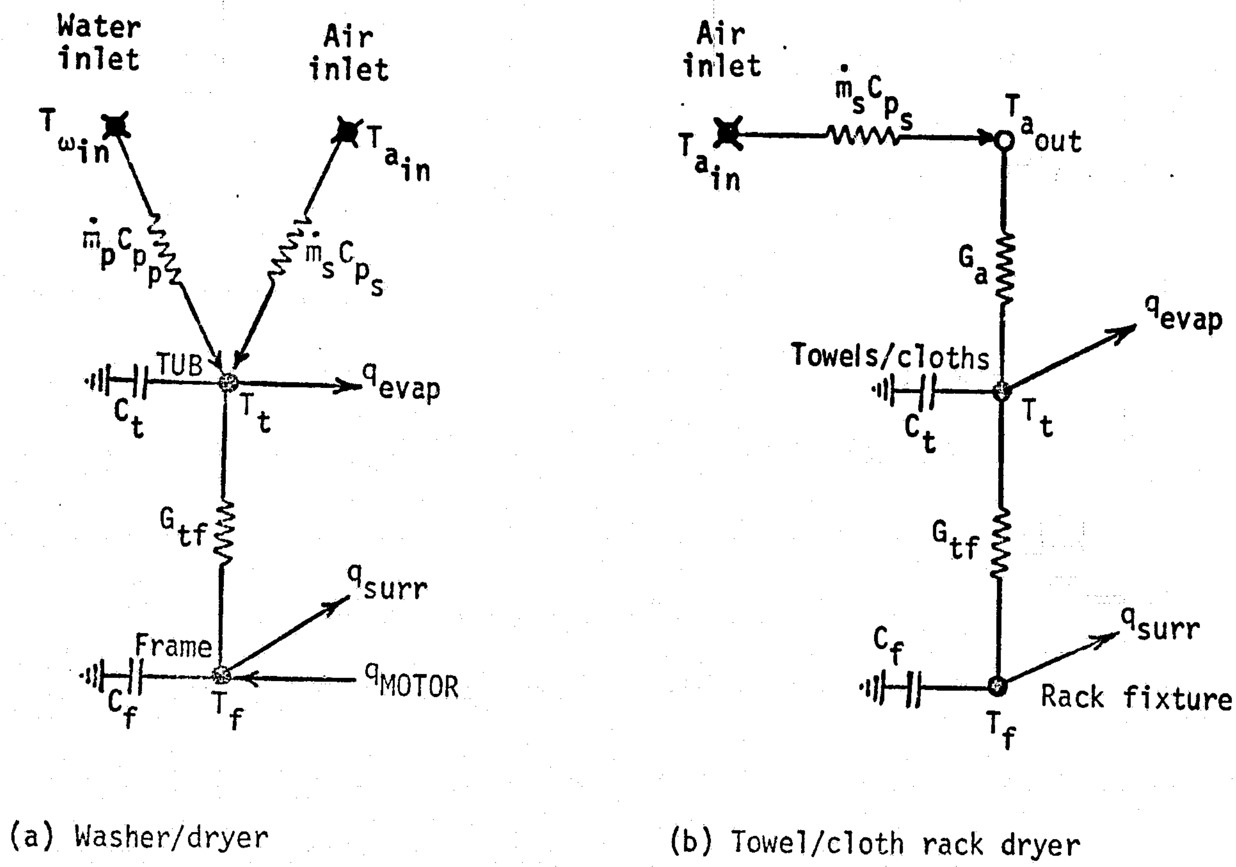


Figure 3-13. Thermal Model for WASDRY Component

3.5.3.1 (Continued)

The latent heat of evaporation q_{evap} is zero except for the drying phase 7. It is evaluated during drying from convective mass transfer considerations discussed in Paragraph 3.5.3.2. The total motor heat generated is assumed to be simply the electrical power for the current cycle phase which is input. The ambient heat loss to the surroundings q_{surr} is computed using the standard G-189A subroutine QSURR which models the conduction, convection, and radiation thermal exchange with the surrounding environment.

For the steady-state case, the value used for γ in equation (3.5.1) is

$$\gamma_{\text{steady state}} = \frac{0.5}{\text{sum of conductors to ground}}$$

Internal steady-state iterations are made until the nodal temperature change computed is less than 0.01°F or until 20 iterations have been made. For a transient solution, γ is computed as

$$\gamma_{\text{transient}} = \frac{\Delta t}{m\ C_p}$$

where $m\ C_p$ is simply the thermal mass of the node. The computing time increment Δt is taken to be 0.5 times the smaller time increment required for computational stability for the two nodes having thermal mass, where

$$\Delta t_{\text{stability}} = \frac{\text{thermal mass}}{\text{sum of conductors to ground}}.$$

Initial runs with a 90-second computing time step had computational instability problems during evaporative drying. This problem was overcome by further limiting the internal computing time increment to a user input value in R(105) during the drying phase. A default value of 10 seconds is set by the subroutine for R(105) and was found satisfactory for the clothes and dishes dryer cases considered. If the system time step is less than the stability time increment, then the system time step is used directly for Δt. If it is greater, then internal iterations of equation (3.5.1) are repeated until the system time step is reached.

3.5.3.1 (Continued)

Fluid flow

In the case of a washer/dryer, the fluid within the tub is assumed to be at the same temperature as the tub and contents. Thus, the air and water inlet flow conductors in Figure 3-13 are evaluated as $\dot{m}\ C_p$. For a towel rack dryer, a convective coefficient is computed. The Reynolds number is evaluated as

$$Re = \frac{\hat{V} L \rho}{\mu} \qquad (3.5.2)$$

The air velocity $\hat{V}$ is input. From the rack surface area A_r, also input, an effective average flow length L is computed assuming a square surface:

$$L = \tfrac{1}{2}\sqrt{A_r} \qquad (3.5.3)$$

Thus, the Reynolds number for flow over a plane surface is computed, with the fluid properties μ and ρ evaluated from the inlet conditions. The convection heat transfer coefficient, assuming laminar flow ($Re<5x10^5$), is then found from the relation:

$$h_c = 0.664 \frac{k_a}{L} Re^{.5}\ Pr^{.33} \qquad (3.5.4)$$

The air outlet temperature is then computed as

$$T_{a_{out}} = \frac{\dot{m}_s\ C_{p_s}\ T_{a_{in}} + G_a\ T_t}{\dot{m}_s\ C_{p_s} + G_a} \qquad (3.5.5)$$

where the convection conductor G_a is the coefficient h_c times the cloth surface area A_r which is input.

3.5.3.2 Evaporative Drying

During the drying phase 7, the evaporative cooling is determined from mass transfer relations for water vapor in air. For a towel rack dryer, since the convection heat transfer coefficient is known from equation (3.5.4), the mass transfer coefficient may be given by

$$h_D = C_M \left(\frac{h_c}{C_{p_a} \rho_a} \right) \left(\frac{Pr}{Sc} \right)^{0.67} \tag{3.5.6}$$

$$Pr \equiv \frac{C_{p_a} \mu_a}{k_a} \qquad Sc \equiv \frac{\mu_a}{\rho_a D} \tag{3.5.7}$$

For a dryer tub, since the convection coefficient was not used directly, the form of the equation for mass transfer over wetted wall columns was used (References 5, 6):

$$h_D = 0.023\, C_M\, Re^{0.83}\, Sc^{0.44} \tag{3.5.8}$$

The Schmidt number Sc is defined in equation (3.5.7). An effective Reynolds number within the dryer tub is found as follows. First, the effective average air velocity $\vec{V}$ relative to the clothes or dishes load may be input, or it may be found from the drum inlet air flowrate and duct size or the drum rotational speed, according to input instruction NSTR(3). Then, for flow continuity,

$$\dot{m}_a = \rho_a A_a \vec{V} \tag{3.5.9}$$

3.5.3.2 (Continued)

An effective hydraulic diameter of the air flow path within the drum is given by

$$d_H \equiv \sqrt{\frac{4\ A_a}{\pi}} = \sqrt{\frac{4\ \dot{m}_a}{\pi\ \rho_a\ \vec{V}}} \qquad (3.5.10)$$

The effective Reynolds number within the dryer drum may then be found from the relation

$$Re = \frac{\vec{V}\ d_H\ \rho_a}{\mu_a} \qquad (3.5.11)$$

Note that in both the mass transfer equations (3.5.6) and (3.5.8), it is primarily the form of the equation that is most important. It relates the relative effects of variations in inlet flow properties, flow rates, and dryer design conditions. The actual magnitudes of the mass transfer co-efficients thus predicted must be correlated with experimental data for reliable accuracy,and the correlation factor C_M was included for this purpose. In the absence of experimental data with which to correlate the analysis, the subroutine will assume a default value of 0.4 for C_M, which was determined from limited analyses of Space Station clothes and dishes dryer concepts.

The water vapor diffusion coefficient D is input for 14.7 psia and 77°F. To adjust for actual dryer conditions, the following equation from Reference 5 is used (where T and P are in absolute units):

$$D \propto \frac{T^{1.75}}{P} \qquad (3.5.12)$$

With the mass transfer coefficient h_D known, the evaporation rate during the drying process is computed from the relation:

$$\dot{m}_{evap} = \frac{h_D\ A}{R_{H2O}} \left[\frac{P_{\omega_{surface}}}{(T_{surface} + 460)} - \frac{P_{\omega_o}}{(T_o + 460)} \right] \qquad (3.5.13)$$

3.5.3.2 (Continued)

and the latent cooling is given by

$$q_{evap} = 1042\ \dot{m}_{evap}$$

Evaporation rate adjustment for "dry" conditions

When the clothes or dishes load is very "wet," evaporation occurs as from a free-standing water surface, and the drying rate may be accurately predicted from equation (3.5.13). However, as drying continues, a point is reached beyond which the actual evaporation rate is less than the computed value. For dish drying, the reason for this is that the exposed dish area becomes partially dried, and the effective wetted area is less than it was initially with the dishes completely wet. A critical moisture content (W_c) is used to determine the point where some of the dishes surface first begins to become dried. The value of W_c must be determined experimentally and is a subroutine input, although a reasonable default value is assumed if it is not. After the critical moisture content is reached, the wetted surface area, and hence the drying rate, is assumed to vary in direct proportion to the amount of water left on the dishes according to the relation:

$$\dot{m}_{evap_{(W<W_c)}} = \dot{m}_{evap_{(equation\ 3.5.13)}} \left(\frac{W}{W_c}\right) \tag{3.5.14}$$

For a clothes load, as the material becomes dried, capillary action and surface tension effects retard the evaporation process and must be accounted for. In early runs, a relatively recent theory assuming a submerged evaporative interface in a porous material was used (References 9, 10, 11). However, it was found this theory did not apply to such thin materials as cloth fabric. Consequently, a simplified approach was taken, Reference 12, in which the drying phase is divided into two distinct zones. With very wet material, a free-standing water surface is assumed and the evaporation rate computed from equation (3.5.13) is assumed valid. As drying continues, a critical moisture content W_c is reached and a "falling-rate" period begins. During this latter period, the evaporation rate predicted by equation (3.5.13) is adjusted by the same equation (3.5.14) given above for a dishes load.

3.5.3.3 Washer/Dryer Phase Switching Logic

The user may choose any of seven phases in which the washer dryer will operate, as described in Paragraph 3.5.1, by setting the input value of K(16) or R(95). However, if input instruction NSTR(5) is given a value of 1, the subroutine will perform its own phase switching logic for transient runs. The program checks each time step to see if the current phase is complete. If it is, the cycle phase R(95) is set to the next phase. It is up to the user, then, to check the cycle phase R(95) in GPOLY logic each time step and set the proper inlet flow conditions. For example, if it is in phase 1 (wash water fill), an inlet water flow should be set. The criterion used to terminate each phase is listed below:

PHASE 0 (unit off): The mission time desired to turn unit on is input. Up to three times per day may be input in R(99, 100, 101). When the turn-on time is reached, either phase 2 (for washers) or phase 7 (for dryers only or towel/cloth drying rack) is selected. Each turn-on time is automatically repeated every 24 mission hours.

PHASE 1 (wash water fill): The weight of water in the washer drum is compared with the maximum weight to be used for the wash cycle in R(102). When that point is reached, the cycle phase is set to 2.

PHASE 2 (wash-circulate): The beginning time of this cycle is stored. When the time allowed for washing [input in R(97)] is reached, cycle phase 3 is selected.

PHASE 3 (spin dry-wash water out): The residual water left after spin-dry is input and also the total time allowed for spin-dry. These inputs determine the water outlet flow rate. When the water loading in the drum has decreased to the input residual level, cycle phase 4 is selected.

PHASE 4 (rinse water fill): The weight of the water in the washer drum is compared with the maximum weight to be used for the rinse cycle in R(103). When that point is reached, the cycle phase is set to 5.

3.5.3.3 (Continued)

PHASE 5 (rinse-circulate): The beginning time of this cycle is stored. When the time allowed for rinsing [input in R(98)] is reached, cycle phase 6 is selected.

PHASE 6 (spin dry-rinse water out): The residual water to be left after spin-dry is input and also the total time allowed for spin-dry. These inputs determine the water outlet flow rate. When the water loading in the drum has decreased to the input residual level, either cycle phase 0 (for washer only) or 7 (for dryer only or washer/dryer) is selected.

PHASE 7 (dry): Two criteria are used to terminate this cycle phase. First, the maximum time allowed is input in R(79). Secondly, the drying phase will be automatically terminated sooner if the residual water loading is less than the input value in R(78). When this point is reached, phase 0 is first selected, and then the next turn-on time is checked as discussed previously.

3.5.3.4 Pressure Drop Considerations

A clothes or dishes washer or dryer have not yet been designed or built for space application. Thus, component pressure drop data, which will depend on the specific hardware design, cannot yet be specified. The clothes dryer design could be quite different from a conventional household dryer due to an apparent optimum drying time of 4 hours (from Reference 21). Thus, much lower air flow rates may be required. The clothes and dishes washers do not actually impose a pressure drop on the spacecraft water system. This is because there is never a water flow from the water system directly through the washer subsystem. Considering the washer cycle phases described in Paragraph 3.5.1, water enters the washer from the water system during the wash and rinse water fill cycles only. At this time, there is no water flow out of the washer and hence no pressure drop through the washer subsystem. Water from the washer enters the water system only during the spin-dry phase. At that time, the washer water pump imposes only an outlet pressure on the water system.

TABLE 3-16

DEFAULT VALUES FOR WASDRY SUBROUTINE INPUT DATA

K(17)=1

R-Array* Location	Default Value
51	70
54	70
55	3
57	70
58	1
60	70
61	4
68	50
70	150
72	70
76	12
77	0.25
81	8
82	16
83	120
86	0.8
88	1
94	0.01974
97	30
98	30
105	10
110	6

R-Array Location	Default Value: Clothes washer NSTR(4)=0	Default Value: Dishwasher NSTR(4)=1
66	4	15.2
75	0.2	0.25
78	0.05	0.0001
79	240	90
80	3343	10587
87	10	0.45
89	0.4	0.4
96(Phases 4 or 7)	R(77)xR(66)	R(94)xR(66)
102	55	15
103	55	15
106	4	4
107	10	10
109	0.2	0.01

*See Section 3.5.2 for definition of variables.

TABLE 3-17

DEFINITION OF SYMBOLS FOR WASDRY SUBROUTINE DESCRIPTION

A	= Surface area (sq ft)
C	= Thermal mass (Btu/°F)
C_M	= Experimental correlation factor to adjust analytical mass transfer coefficient (dimensionless)
C_p	= Thermal specific heat (Btu/lb °F)
D	= Diffusion coefficient for water vapor in air (sq ft/hr)
d_H	= Effective dryer air flow path hydraulic diameter (ft)
G	= Thermal conductor (Btu/hr °F)
G_{tf}	= Thermal conductor between washer/dryer tub and frame nodes (or between towel/cloths and drying rack fixture); (Btu/hr-°F)
h_c	- Thermal convection heat transfer coefficient of air over dryer contents (Btu/hr-sq ft-°F)
h_D	= Evaporative mass transfer coefficient (ft/hr)
k_a	= Air thermal conductivity (Btu/hr-ft-°F)
L	= Effective average air flow path length through dryer (ft)
$\dot{m}$	= Mass flow rate (lb/hr)
Pr	= Prandtl number for dryer air (dimensionless)
q	= Heat transfer (Btu/hr)
q_{evap}	= Latent heat of evaporation in dryer (Btu/hr)
q_{MOTOR}	= Washer/dryer total motor heat dissipated (Btu/hr)
q_{surr}	= Washer/dryer frame node net heat loss to ambient (Btu/hr)
R	= Ideal gas constant (ft-lb/lb-°R)
Re	= Effective air Reynolds number (dimensionless)
Sc	= Schmidt number for dryer air (dimensionless)
p	= Pressure (psia)

TABLE 3-17 (concluded)

DEFINITION OF SYMBOLS FOR WASDRY SUBROUTINE DESCRIPTION

T	= Temperature ($^\circ$F, except where otherwise noted)
Δt	= Transient solution time increment (hours)
$\vec{V}$	= Effective air velocity relative to dryer contents (ft/hr)
ρ	= Dryer air density (lb/cu ft)
μ	= Dryer air viscosity (lb/hr-ft)
γ	= Multiplier to compute new temperatures, defined by equation (3.5.1); (dimensionless)
W	= Moisture content in washer/dryer load (lb water/lb dry clothes or dishes)
W_c	= Critical moisture content below which evaporation rate is less than predicted rate for free-standing water surface

Subscripts

a	= Dryer air
f	= Washer/dryer frame node (or towel drying rack node)
H2O	= Water
in	= Inlet condition
new	= Condition at end of current time step or iteration
o	= Average dryer air condition away from cloth or dishes surface
old	= Condition at beginning of current time step or iteration
p	= Primary side (water)
r	= Drying rack
s	= Secondary side (air)
t	= Washer/dryer tub and contents node (or towels/cloths node for rack dryer)
ω	= Water

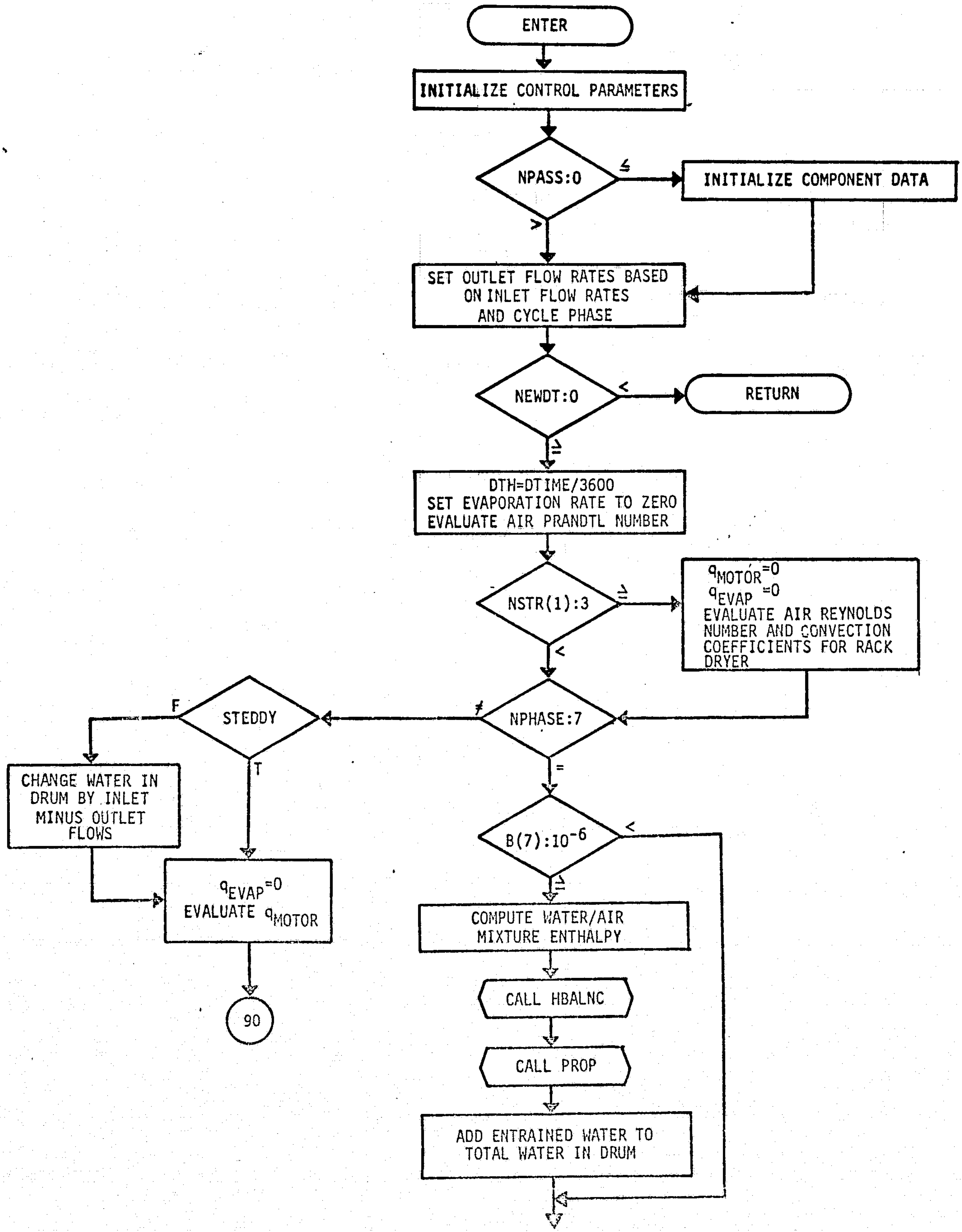


Figure 3-14. Logic Flow Chart for WASDRY Subroutine

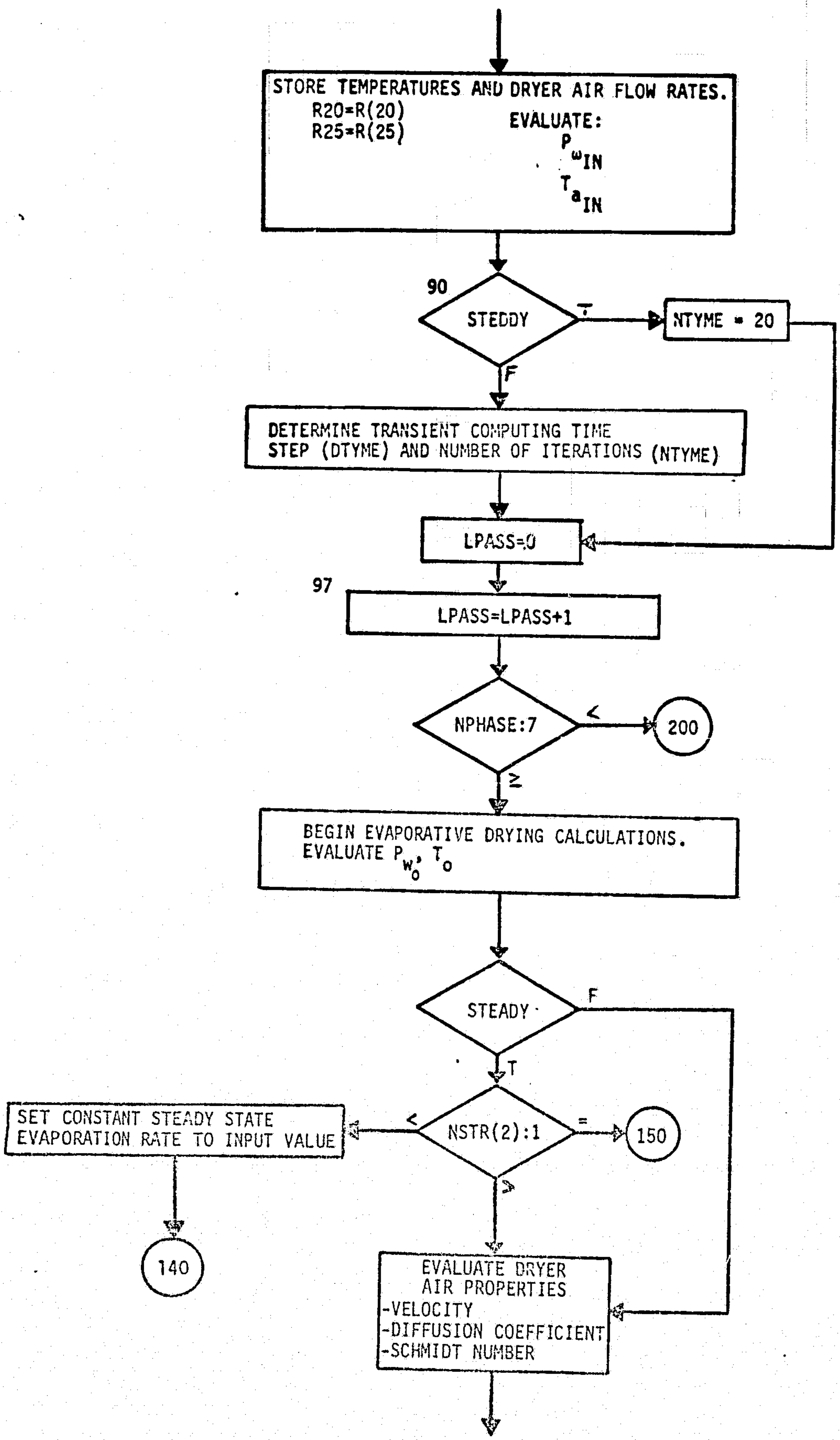


Figure 3-14. Logic Flow Chart for WASDRY Subroutine (Continued)

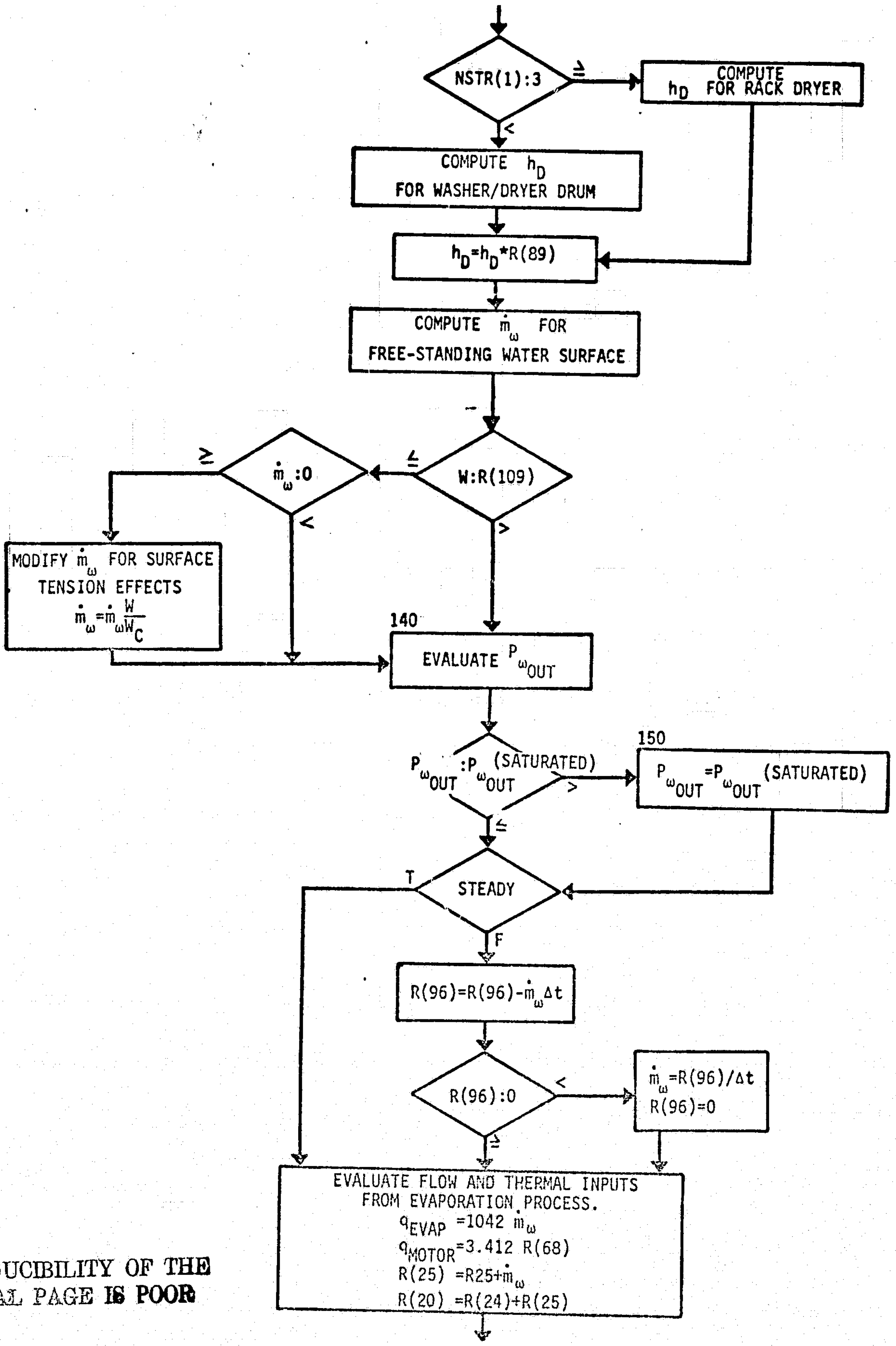


Figure 3-14. Logic Flow Chart for WASDRY Subroutine (Continued)

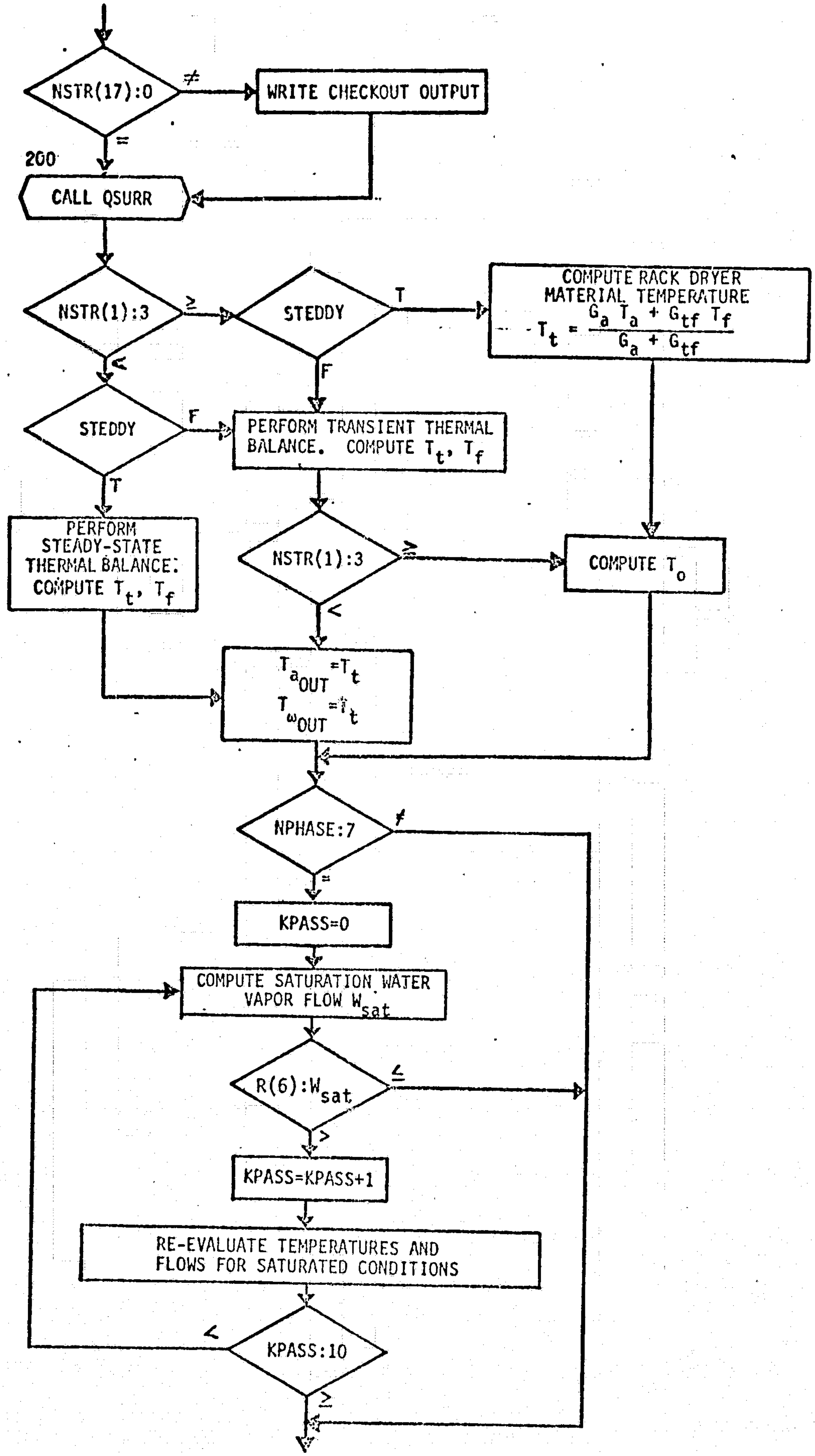


Figure 3-14. Logic Flow Chart for WASDRY Subroutine (Continued)

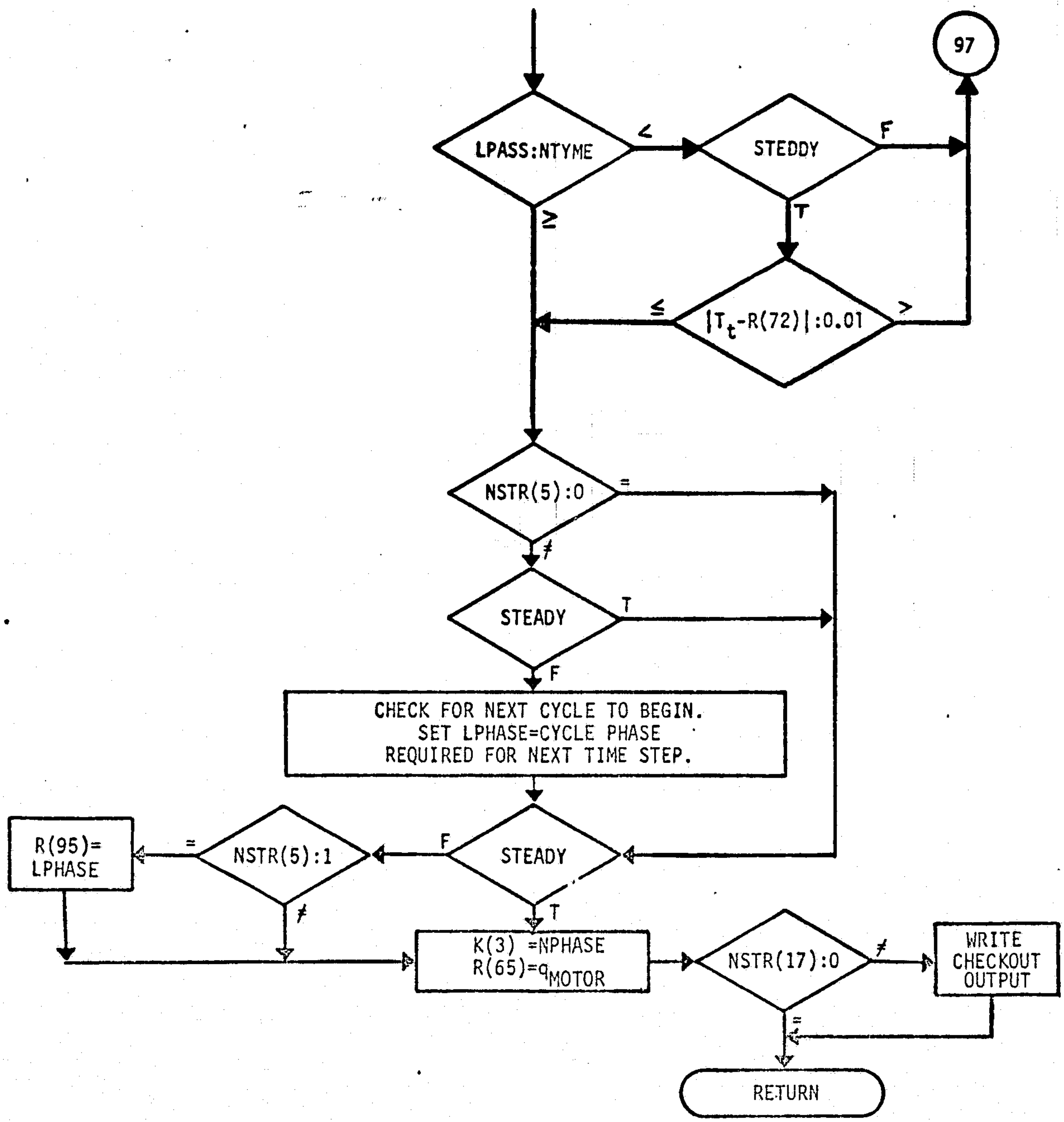


Figure 3-14. Logic Flow Chart for WASDRY Subroutine (Concluded)

TABLE 3-18

DEFINITION OF WASDRY SUBROUTINE VARIABLES

C	=	Combined thermal mass of washer drum, water and load (Btu/°F)
CSTART	=	Start time of current cycle phase (seconds)
DH	=	Effective hydraulic diameter of air flow path inside dryer drum (ft)
DIFF	=	Water vapor diffusion coefficient in air (sq ft/hr)
DTH	=	System transient time step (hours)
DTYME	=	Internal transient computing time step used (hours)
EVAP	=	Water evaporation rate (lb/hr)
GAIR	=	Thermal conductor from washer load to air inlet (Btu/hr-°F)
HC	=	Heat transfer coefficient of air over clothes on drying rack (Btu/hr-sq ft-°F)
HD	=	Mass transfer coefficient for water evaporation (ft/hr)
HMIX	=	Total enthalpy of dryer inlet air (Btu/hr)
KPASS	=	Iteration counter for determining saturated conditions
LPASS	=	Internal heat and mass balance iteration counter
LPHASE	=	Variable used to select washer phase for next iteration
NPHASE	=	Operational cycle phase assumed (0 through 7)
NTYME	=	Number of internal transient computing iterations
PR	=	Air Prandtl number (dimensionless)
PVAP	=	Computed outlet water vapor pressure (psia)
PWIN	=	Inlet air water vapor pressure (psia)
PWINF	=	Effective water vapor pressure in drum (or over drying rack); (psia)
PWOUT	=	Outlet water vapor pressure on previous iteration (psia)

TABLE 3-18 (concluded)

DEFINITION OF WASDRY SUBROUTINE VARIABLES

QEVAP = Latent heat of evaporation (Btu/hr)

QMOTOR = Heat added to structure from drum motor (Btu/hr)

R20,R25 = Variables used to store value of R(20) and R(25) at beginning of system iteration

RE = Air Reynolds number (dimensionless)

RESID = Residual water retention required to terminate drying process (lb water/lb dry clothes or dishes)

SC = Air Schmidt number (dimensionless)

TTUB = Variable used to store washer load temperature prior to making thermal balance (°F)

TWIN = Inlet air temperature (°F)

TWINF = Effective air temperature in drum (or over drying rack); (°F)

TWOUT = Outlet air temperature on previous iteration (°F)

VMAX = Maximum absolute velocity of air over clothes or dishes (ft/hr)

VREL = Air velocity relative to clothes or dishes (ft/hr)

W = Water retention in load (lb water/lb dry clothes or dishes)

3.6 WASTEC

The WASTEC subroutine is designated as G-189A No. 68.

3.6.1 Subroutine Description

This subroutine will simulate a urine/fecal waste collector applicable to space use, such as a urinal or dryjohn. A flow schematic of the WASTEC component is shown in Figure 3-15. Three operational usage phases may be simulated:

Phase 0 - Unit off
Phase 1 - Urine collection
Phase 2 - Fecal collection
Phase 3 - Combined urine/fecal collection

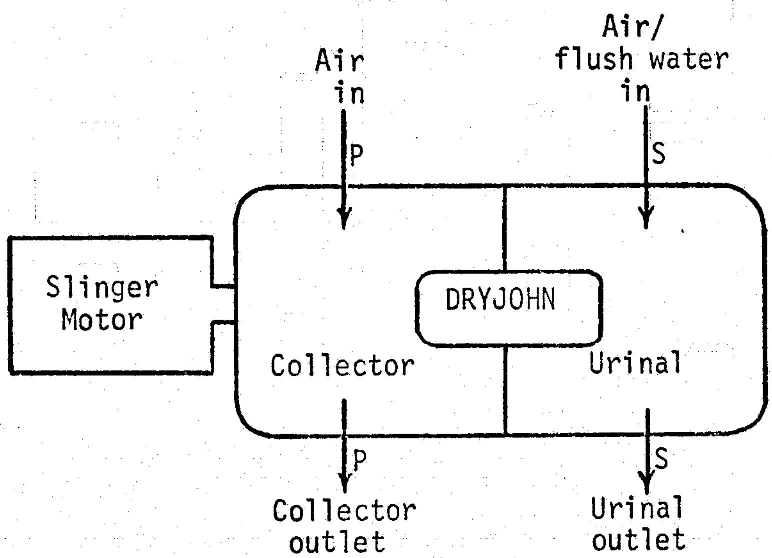


Figure 3-15. WASTEC Component Flow Schematic

Commode contents may be under vacuum, if requested, during phases 0 and 1. Thermal exchange between the collector and ambient environment is modeled using the standard G-189A library routine QSURR. The subroutine

3.6.1 (Continued)

models the thermal/mass exchange in the collector and urinal only, with the peripheral pumps, valves, etc. to be simulated by standard G-189A component routines.

3.6.2 Subroutine Data

3.6.2.1 General Notes

a. The flow paths for the collector and urinal are isolated. The collector is on the component primary side, and the urinal is on the secondary.

b. All gaseous flow codes (1, 2, and 3) are permitted for both the collector and urinal. For a urinal flush, the water must be inlet as entrained liquid in a gaseous fluid.

c. Vacuum exhaust and drying of the collector are optional according to input instruction NSTR(1). If NSTR(1)=0, a collector vacuum is automatically assumed during phases 0 and 1. Each time vacuum is initiated following operation of a fecal collection phase (2 and 3), a pumpdown or blowdown is automatically performed. The time required for this is input. The vacuum pump, if used, is not a part of the WASTEC component.

d. To satisfy the total cabin heat balance, the heat lost through the waste collector structure, R(53), should be added to the associated cabin component sensible heat addition [cabin component location R(66)] in GPOLY logic each iteration.

3.6.2.2 Instruction Options

NSTR(1): Controls vacuum drying of commode contents

= 0 Commode contents are assumed to be under vacuum continually except during fecal collection (phases 2 and 3). Vacuum pumpdown is activated automatically (during transient run only) immediately after fecal collection or at beginning of run.

= 1 Commode contents are assumed at ambient pressure.

3.6.2.3 Heat Loss V-Array Data

Reference Location	Description	Data Type
51	Collector contents temperature (°F)	I(0), 0
52	Summed thermal conductance between collector contents and ambient, excluding conductance to air flow through collector or urinal (Btu/hr °F)	0
53	Collector contents net heat loss to ambient, excluding heat transfer to air flow through collector and urinal (Btu/hr)	0
54	Ambient gas temperature (°F)	I(0)
55	Thermal convection conductor from collector outer surface to ambient gas (Btu/hr °F)	I(0)
56	Convective heat loss from collector outer surface to ambient gas (Btu/hr)	0
57	Ambient wall temperature (°F)	I(0)
58	Thermal radiation conductor (A℈) between ambient walls and collector outer surface (sq ft)	I(0)
59	Radiative heat loss from collector outer surface to ambient walls (Btu/hr)	0
60	Temperature of ambient structure connected to collector (°F)	I(0)
61	Thermal conductor from collector node to attached ambient structure (Btu/hr °F)	I(0)
62	Thermal loss from collector to attached ambient structure (Btu/hr)	0
63	Collector outer surface temperature (°F)	0
64	Thermal conductor between collector contents and collector outer surface (Btu/hr °F)	I(0)

3.6.2.4 Steady-State V-Array Data

Reference Location	Description	Data Type
65	Net heat generated in unit by slinger motor (Btu/hr)	0
66	Operational usage phase (0 through 4 as described previously)	I(R)
67	Slinger motor rated electrical power (watts)	I(0)
68	Not used	
69	Urine flow rate during micturition (lb/hr)	I(0)
70	Not used	
71	Not used	
72	Not used	
73	Not used	
74	Exposed surface area of collector contents, used for evaporative drying calculations (sq ft)	I(0) if NSTR(1)=0
75	Evaporation coefficient f in equation (3.6.6) for vacuum drying (dimensionless)	I(R) if NSTR(1)=0
76	Evaporation or sublimation rate of collector contents computed from equation (3.6.6); (lb/hr)	0
77	Effective thermal conductor between air flow through collector and collector contents (Btu/hr °F)	I(0)
78	Effective thermal conductor between flow through urinal and collector contents (Btu/hr °F)	I(0)
79	Urine and feces inlet temperature (°F)	I(0)

3.6.2.5 Transient K-Array Data

Reference Location	Description	Data Type
16	Operational cycle phase used on previous time step	0
17	This variable is used in certain program logic to indicate pressure inside commode during the previous time step, as follows: - 1 → ambient pressure 0 → hard vacuum 5 → vacuum pumpdown or blowdown applied	0

3.6.2.6 Transient V-Array Data

Reference Location	Description	Data Type
80	Time required for vacuum pumpdown or blowdown (min)	I(0) if NSTR(1)=0
81	Internal empty volume of fecal collector (cu ft)	I(0) if NSTR(1)=0
82	Total cumulative mass of urine collected during current usage phase (lb)	0
83	Total urine generated per crewman (lb/use)	I(0)
84	Total mass of free gas in collector to be evacuated during vacuum pumpdown (lb)	0
85	Effective thermal mass of empty collector node (Btu/°F)	I(0)
86	Total fecal mass added to collector per usage (lb)	I(0)
87	Inlet fecal moisture content (lb H_2O/ lb wet fecal material)	I(0)
88	Specific heat of dried feces (Btu/lb °F)	I(0)

3.6.2.6 (Continued)

Reference Location	Description	Data Type
89	Moisture present in collector in liquid phase (lb)	I(0), 0
90	Moisture present in collector in solid phase (lb)	I(0), 0
91	Total dry mass of collector contents (lb)	I(0), 0
92	Not used	
93	Start time of current drying phase (seconds)	0
94-98	Not used	

3.6.2.7 Extra V-Array Data (must be specified for a transient case)

Reference Location	Description	Data Type
99	Reserved to store density (lb/cu ft) of upstream primary source. Used for initial condition for vacuum pumpdown.	0
100→x	Reserved to store R-array flow data for inlet primary flow on previous iteration. Used to store initial flow and constituents for vacuum pumpdown. Reserve 4+5*NPF locations.	0

3.6.3 Analytical Model Description

3.6.3.1 Thermal Balance

The thermal model assumed for the WASTEC component is shown in Figure 3-16. The collector contents temperature is computed by a standard forward-difference method:

$$T_{c_{new}} = T_{c_{old}} + \gamma \, \Sigma q_{into\ node} \tag{3.6.1}$$

3.6.3.1 (Continued)

$$\Sigma q_{\text{into node}} = G_c \left(T_{p_{out}} - T_c\right) + G_u \left(T_{s_{out}} - T_c\right) + q_{MOTOR} - q_{surr} - q_{evap} \quad (3.6.2)$$

The total slinger motor heat generated is assumed to be simply the electrical input power for the phase being simulated. The latent heat of evaporation or sublimation q_{evap} is zero except when the collector is being vacuum dried. During vacuum drying, the latent heat is computed as discussed in Paragraph 3.6.3.2. The ambient heat loss to the surroundings q_{surr} is computed using the standard G-189A subroutine QSURR which models the conduction, convection and radiation thermal exchange with the surrounding environment.

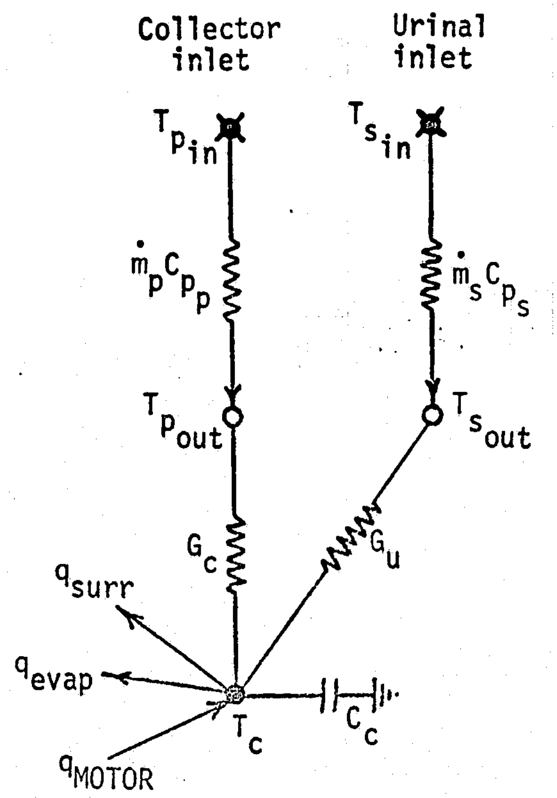


Figure 3-16. Thermal Model for WASTEC Component

For the steady-state case, the value used for γ in equation (3.6.1) is

$$\gamma_{\text{steady state}} = \frac{0.5}{\text{sum of conductors to ground}}$$

Internal steady-state iterations are made until the nodal temperature change computed is less than 0.05°F or until 25 iterations have been made. For a transient solution, γ is computed as

$$\gamma_{\text{transient}} = \frac{\Delta t}{C_c}$$

3.6.3.1 (Continued)

The computing time increment Δt is taken to be 0.4 times the time increment required for computational stability, where

$$\Delta t_{stability} = \frac{C_c}{\text{sum of conductors to ground}}$$

If the system time step is less than the stability time increment, the system time step is used directly for Δt. If it is greater, then internal iterations of equation (3.6.1) are repeated until the system time step is reached. After computing the collector contents temperature T_c, the fluid outlet temperatures are calculated as follows:

$$T_{p_{out}} = \frac{\dot{m}_p C_{p_p} T_{p_{in}} + G_c T_c}{\dot{m}_p C_{p_p} + G_c} \tag{3.6.3}$$

$$T_{s_{out}} = \frac{\dot{m}_s C_{p_s} T_{s_{in}} + G_u T_c}{\dot{m}_s C_{p_s} + G_u} \tag{3.6.4}$$

3.6.3.2 Vacuum Drying

During vacuum drying of the collector, the evaporation or sublimation rate is assumed to be governed by the Hertz-Knudsen relation (References 13, 14):

$$\dot{m}_\omega = 2.784 \, \alpha \, A_c \left(P_{\omega_{surface}} - P_\infty \right) \sqrt{\frac{M_\omega}{2 \pi R T_c}} \tag{3.6.5}$$

where pressures are in mm Hg and the temperature is in °R. The accommodation coefficient α has a theoretical maximum value of 1, but varies widely with the purity of the liquid being evaporated and the substrate material (References 15, 16, 17). A discussion of the application of equation (3.6.5)

3.6.3.2 (Continued)

to freeze-drying of foods is given in Reference 18. In standard freeze-drying practice (Reference 13), the gas pressure P_∞ above the evaporating or sublimating surface is assumed negligible compared with the water vapor partial pressure $P_{\omega_{surface}}$, and the water loss rate is given by:

$$\dot{m}_\omega = f_\omega A_c P_{\omega_{surface}} \sqrt{\frac{M_\omega}{2 \pi R T_c}}$$

Since essentially only water is being evaporated in the collector, the constants M_ω, π, R, and f_ω were combined into one constant f, and the following equation

$$\dot{m}_\omega = \frac{f A_c P_{\omega_{surface}}}{\sqrt{T_c}} \qquad (3.6.6)$$

was used in the subroutine to compute the evaporation or sublimation rate of the moisture in the collector contents, where $P_{\omega_{surface}}$ is in psia. The evaporation coefficient f is a subroutine input and may be varied to correlate with experimental data. A default value for f of 3.0 is used in the subroutine, which was found to agree best with limited test results. The outlet flow rate is assumed to be saturated water vapor, and the G-189A subroutine PSAT is used to compute the outlet gas pressure. The latent cooling is given by:

$$q_{evap} = \begin{cases} 1042\ \dot{m}_\omega \text{ (evaporation, } T_c \geq 32°\text{F)} \\ 1225\ \dot{m}_\omega \text{ (sublimation, } T_c < 32°\text{F)} \end{cases} \qquad (3.6.7)$$

During a transient solution, the water/ice balance is kept track of. When passing through the freezing point, the collector contents temperature is held constant at 32°F until all the water has either melted or frozen.

3.6.3.3 Mass Flow Additions

During phases 1 (urine collection), 2 (fecal collection), and 3 (urine/fecal collection), mass is added to the dryjohn and/or air flow. During phases 1 and 3, the urine is added as entrained water in the inlet air flow according to the micturition flow rate which is input. The total void mass per man is also input, and when the total urine flow has reached this amount it is terminated for the current phase. It will automatically begin again when phase 1 or 3 is reinitiated. Each time phase 2 or 3 is initiated, the collector contents mass and water contents are increased by the total defecation mass per crewman, which is input.

3.6.3.4 Pressure Drop Considerations

The current dryjohn subsystem, which will be flown on Shuttle, is still in the design stage; and pressure drop data for the various components are not yet available. Consequently, the G-189A pressure drop model input data cannot be specified at this time. However, when the data become available, the method described previously for the shower data [depicted by equation (3.4.33)] may be used. One possible exception to this is during urine collection, when there can be two-phase flow of urine mixed with air. For this condition the pressure between the urinal and water separator may fluctuate, and an attempt to include this in the model could cause computational instability problems.

TABLE 3-19

DEFAULT VALUES FOR WASTEC SUBROUTINE INPUT DATA

R-Array* Location	Default Value
51	60
54	70
55	2
57	70
58	0.8
60	70
61	1
64	2.5
67	20
69	108
74	1
75	3
77	1.5
78	0.15
79	97
80	3
81	1.2
83	0.6
85	0.5
86	0.3
87	0.75
88	0.25
89 + 90	.08 R(91)
91	2.5

*See Section 3.6.2 for definition of variables.

TABLE 3-20

DEFINITION OF SYMBOLS FOR WASTEC SUBROUTINE DESCRIPTION

A_c = Exposed surface area of collector contents (sq ft)

C = Nodal thermal mass (Btu/°F)

C_p = Thermal specific heat (Btu/lb°F)

f = R(75) = Modified vacuum evaporation coefficient used in subroutine, equation (3.6.6)

f_ω = Evaporation/sublimation coefficient for vacuum drying (dimensionless)

G_c = R(77) = Effective thermal conductor between air flow through collector and collector contents (Btu/hr °F)

G_u = R(78) = Effective thermal conductor between flow through urinal and collector contents (Btu/hr °F)

$\dot{m}$ = Mass flow rate (lb/hr)

M = Molecular weight

P_ω = Water vapor partial pressure (psia unless otherwise noted)

P_∞ = Gas pressure above collector surface (mm Hg)

q = Heat transfer rate (Btu/hr)

q_{evap} = Latent heat of evaporation or sublimation in collector (Btu/hr)

q_{MOTOR} = Total heat input to collector from motor (Btu/hr)

q_{surr} = Collector node net heat loss to ambient (Btu/hr)

R = Universal gas constant (ft lb/mole-°R)

T = Temperature (°F unless otherwise noted)

Δt = Transient solution time increment (hours)

γ = Multiplier to compute new temperatures, defined by equation (3.6.1); (dimensionless)

α = Accommodation coefficient for vacuum evaporation/sublimation rate, equation (3.6.5); (dimensionless)

TABLE 3-20 (concluded)

DEFINITION OF SYMBOLS FOR WASTEC SUBROUTINE DESCRIPTION

Subscripts

c = Collector

in = Inlet flow condition

new = Condition at end of current time step or iteration

old = Condition at beginning of current time step or iteration

out = Outlet flow condition

p = Primary side (collector)

s = Secondary side (urinal)

u = Urinal

ω = water

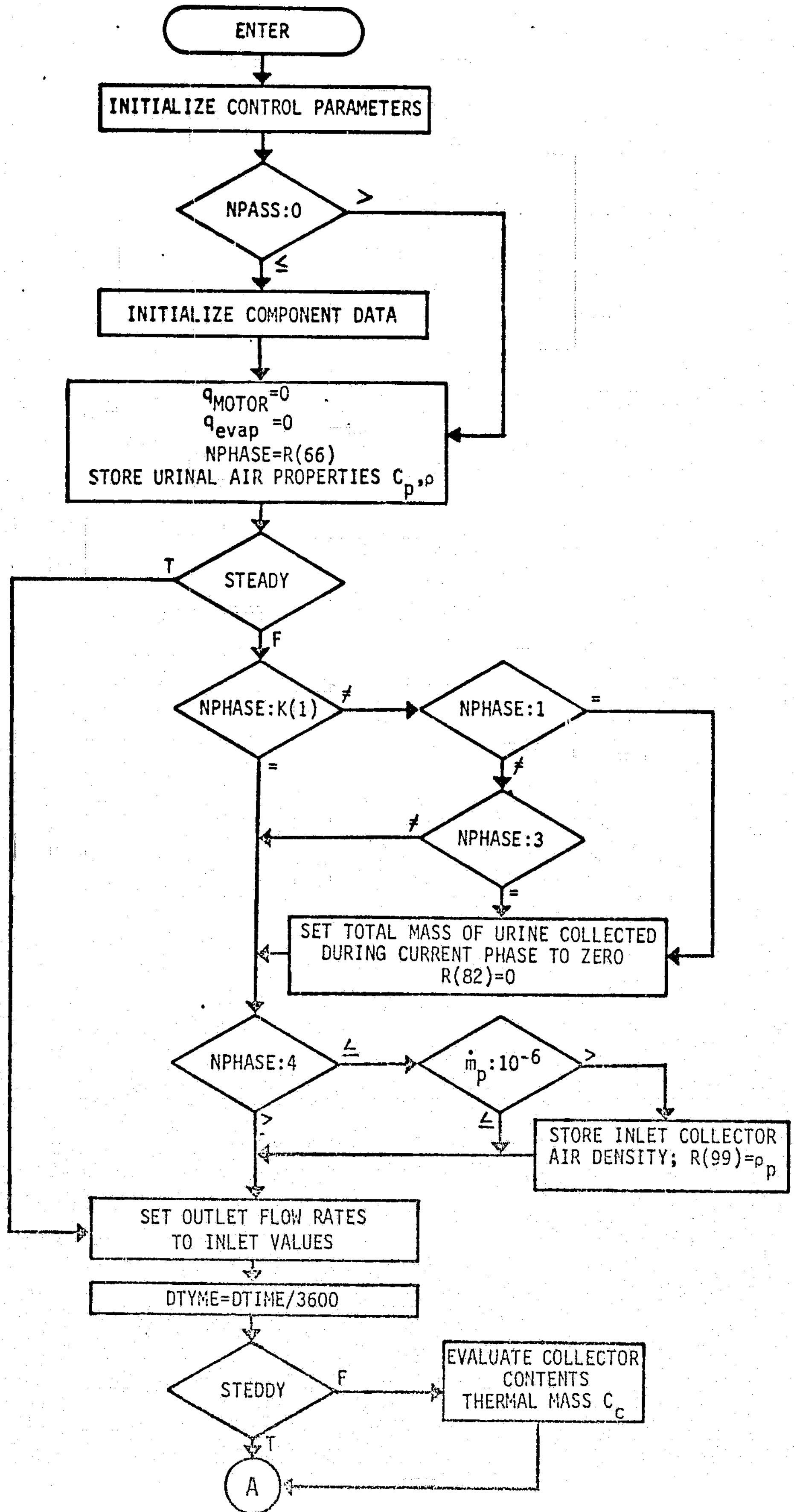


Figure 3-17. Logic Flow Chart for WASTEC Subroutine

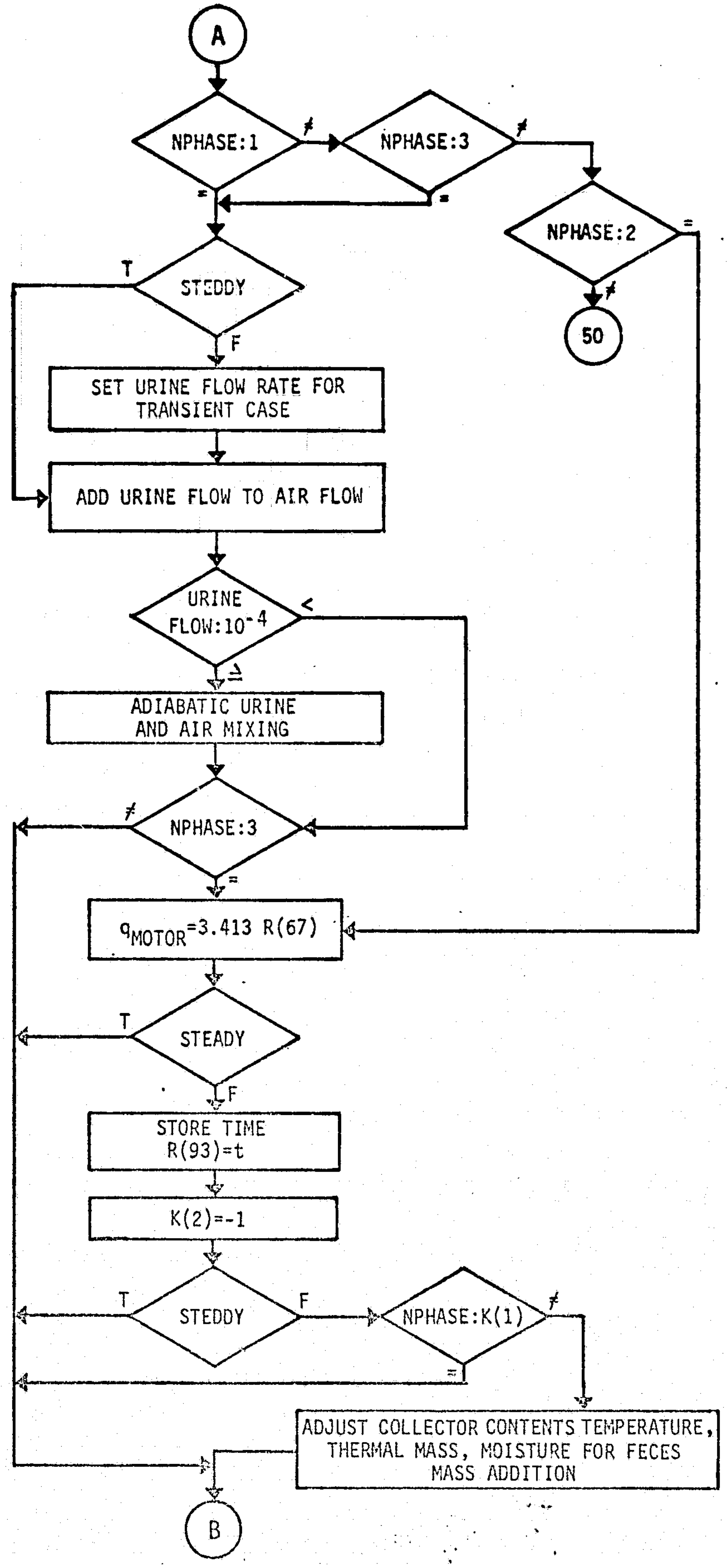


Figure 3-17. Logic Flow Chart for WASTEC Subroutine (Continued)

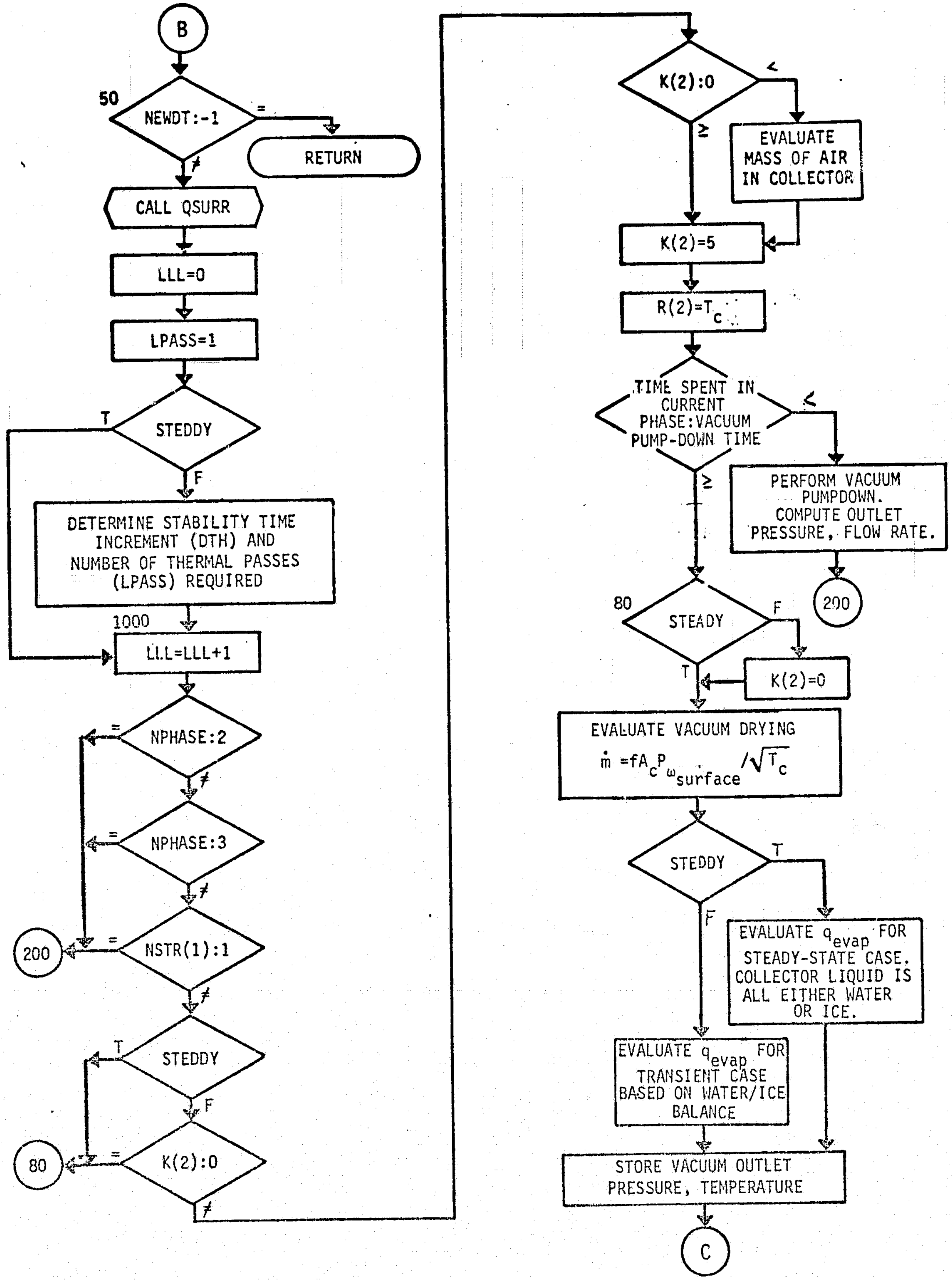


Figure 3-17. Logic Flow Chart for WASTEC Subroutine (Continued)

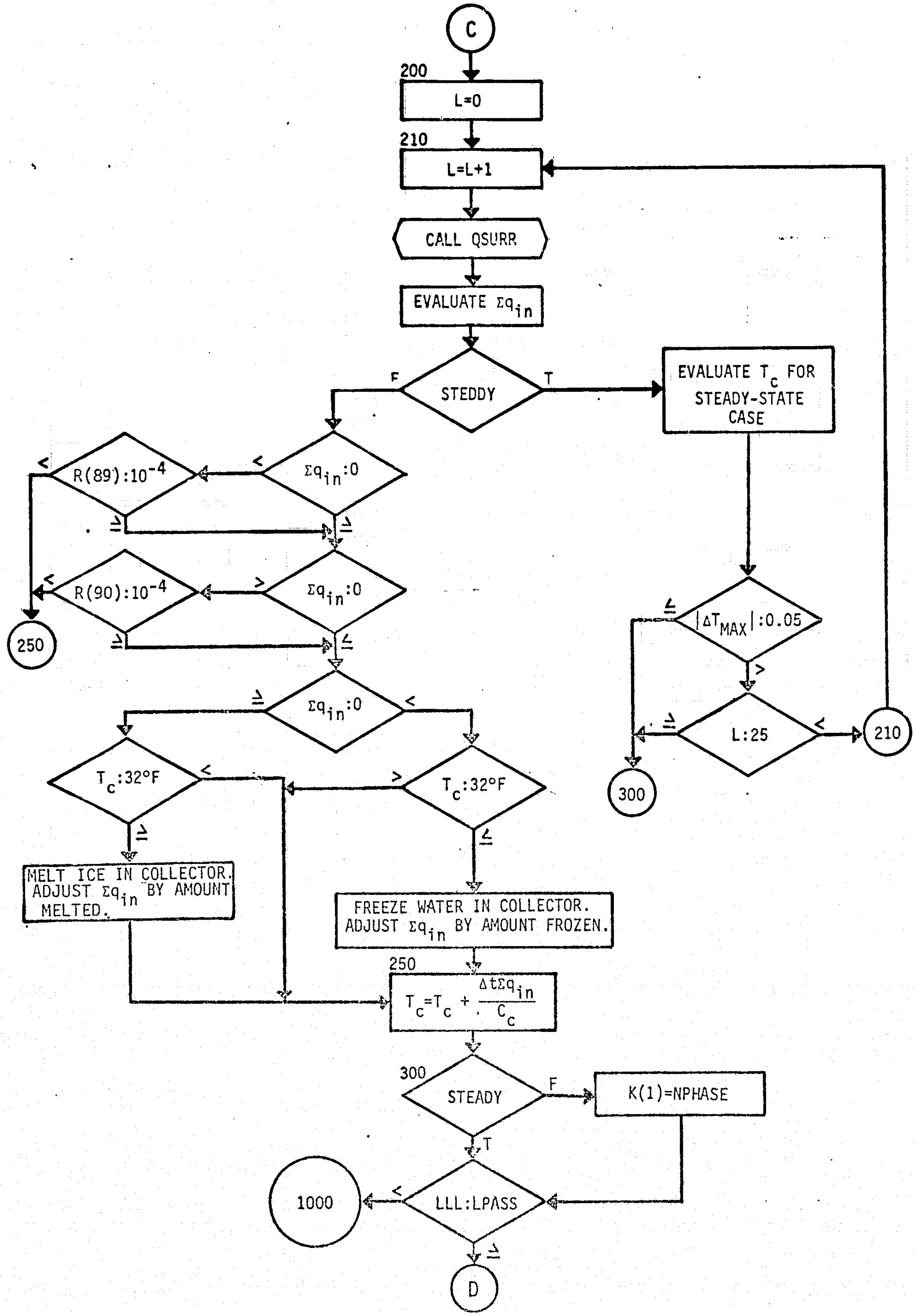


Figure 3-17. Logic Flow Chart for WASTEC Subroutine (Continued)

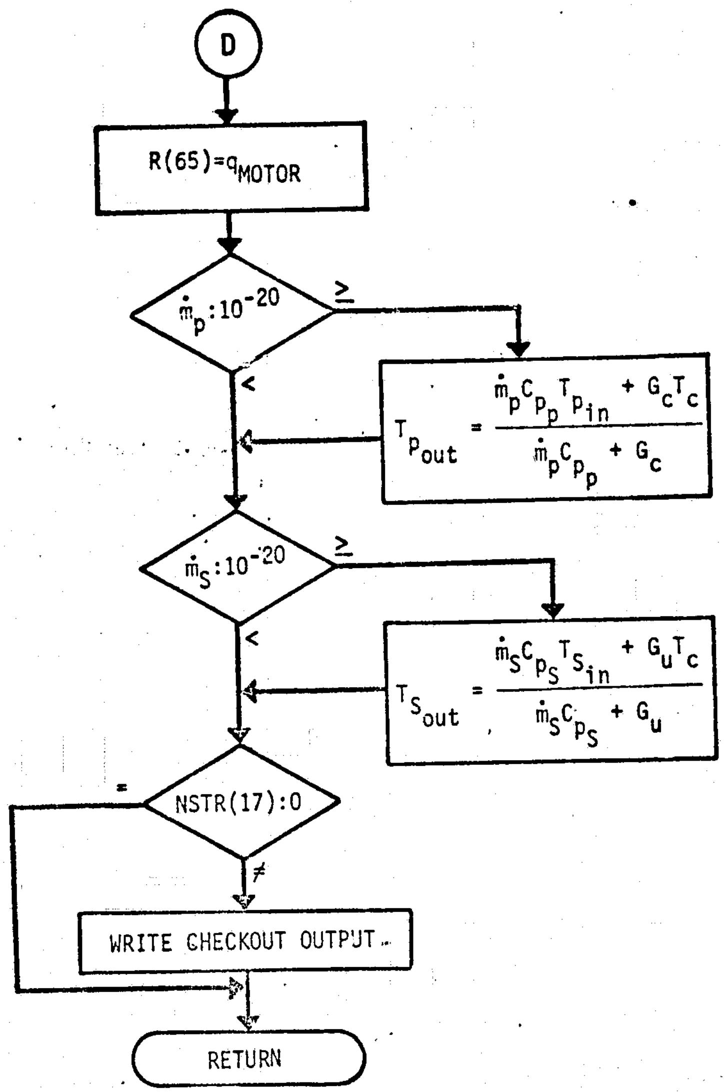


Figure 3-17. Logic Flow Chart for WASTEC Subroutine (Concluded)

TABLE 3-21

DEFINITION OF WASTEC SUBROUTINE VARIABLES

CAP	=	Thermal capacitance of collector and contents (Btu/°F)
CPBB	=	Variable used to store value of inlet air property CPB
DTMIN	=	Time elapsed from start of vacuum pumpdown (minutes)
DTYME	=	System transient time step (hours)
HMIX	=	Total enthalpy of inlet urinal air mixture (Btu/hr)
LLL	=	Internal heat balance iteration counter
LPASS	=	Number of heat balance iterations required
NPHASE	=	Operational usage phase assumed (0 through 3)
QEVAP	=	Latent heat of evaporation or sublimation of collector contents moisture (Btu/hr)
QIN	=	Net heat transfer into collector and contents (Btu/hr)
QMOTOR	=	Heat added to collector from slinger motor (Btu/hr)
RHOBB	=	Variable used to store value of inlet air property RHOB
SUBEVA	=	Vacuum evaporation or sublimation rate (lb/hr)
UFLOW	=	Urine flow rate addition (lb/hr)

4.0 SUBROUTINE MODEL VERIFICATION RESULTS

Each of the appliance subroutines was initially checked for accuracy by modeling a single G-189A component and running it separately (i.e., not connected in an all-up system model) with dummy ambient and inlet flow conditions. These conditions were held constant during a run and a steady-state and transient solution obtained. The results of these checkout runs are described in the following paragraphs for each appliance subroutine. Calculations are presented to verify conservation of energy within the components, and the solutions are compared with test data where available. Many of these appliances have not yet been built for spacecraft application, and available test data required for correlation of the models are limited. For such cases, additional testing and model correlation are recommended.

The inlet flow conditions tabulated in the following paragraphs for each component consist of the standard G-189A A- and B-arrays. The A-array contains the primary side inlet flow rates, temperatures, and pressures; the B-array contains the same data for the secondary side. The definition of these arrays is given in Table 4-1, which is taken from the G-189A Program Manual (Reference 2). In addition, the R-array locations 1 through 38, which have the same definition for all components, are described in Table 4-1. These R-array locations comprise the outlet flow conditions for the component solutions given in the following paragraphs. For a more complete description of these arrays, the G-189A Program Manual (Reference 2) should be consulted. The rest of the component solutions, as well as their model input data, are stored in the remainder of the R-array; and these R-array locations for each component are described in the subroutine descriptions in Section 3.

TABLE 4-1

G-189A COMPONENT INLET AND OUTLET FLOW DATA DEFINITION

Primary Source Flow Data	Secondary Source Flow Data	Primary Side Component Data	Secondary Side Component Data		Liquid Flows (Flow Code = 0, 4)	Gaseous Flows (Flow Code = 1, 2, 3)
A(1)	B(1)	R(1)	R(20)	Total Flow (lb/hr)	0, 4	1, 2, 3
A(2)	B(2)	R(2)	R(21)	Fluid Temperature (°F)	0, 4	1, 2, 3
A(3)	B(3)	R(3)	R(22)	Upstream Duct Outlet Pressure (psia)	0, 4	1, 2, 3
A(4)	B(4)	R(4)	R(23)	Component Outlet Pressure (psia)	0, 4	1, 2, 3
A(5)	B(5)	R(5)	R(24)	Non-Condensables Flow (lb/hr)		1, 2, 3
A(6)	B(6)	R(6)	R(25)	Condensable Vapor Flow (lb/hr)		1, 2, 3
A(7)	B(7)	R(7)	R(26)	Condensable Entrained Liquid Flow (lb/hr)		1, 2, 3
A(8)	B(8)	R(8)	R(27)	Non-Condensables Specific Heat (Btu/lb-°F)		1, 2, 3
A(9)	B(9)	R(9)	R(28)	Non-Condensables Molecular Weight (lb/mol)		1, 2, 3
A(10)	B(10)	R(10)	R(29)	Oxygen Flow (lb/hr)		2, 3
A(11)	B(11)	R(11)	R(30)	Diluent Flow (lb/hr)		2, 3
A(12)	B(12)	R(12)	R(31)	Carbon Dioxide Flow (lb/hr)		2, 3
A(13)	B(13)	R(13)	R(32)	Trace Contaminants Flow (lb/hr)		2, 3
A(14)	B(14)	R(14)	R(33)	Special Flow No. 1 (lb/hr)		2, 3
A(15)	B(15)	R(15)	R(34)	Special Flow No. 2 (lb/hr)	4	3
A(16)	B(16)	R(16)	R(35)	Special Flow No. 3 (lb/hr)	4	3
A(17)	B(17)	R(17)	R(36)	Special Flow No. 4 (lb/hr)	4	3
A(18)	B(18)	R(18)	R(37)	Special Flow No. 5 (lb/hr)	4	3
A(19)	B(19)	R(19)	R(38)	Special Flow No. 6 (lb/hr)	4	3

4.1 CHILLR CHECKOUT RUN

The CHILLR subroutine was used to model a refrigerator and freezer of the approximate size of those used on Skylab. For the refrigerator simulation, an externally chilled Freon-21 coolant was routed through cooling tubes embedded in the refrigerator walls. For the freezer simulation, a self-contained cooling unit was assumed with a constant coefficient of performance of 1.07. This value represents a feasible vapor-compression refrigeration unit once considered for use on Shuttle (Reference 22). For both cases a steady-state and transient solution were obtained, and the results presented in Sections 4.1.1 through 4.1.4. In addition, a CHILLR run simulating the Stirling cycle freezer designed for Shuttle (Reference 32) is described in Section 4.1.5.

4.1.1 CHILLR Steady-State Refrigerator Case

The G-189A component input data for the refrigerator case are listed in Table 4-2. The final steady-state solution is tabulated in Table 4-3. The results for the run are plotted in Figures 4-1 and 4-2. As most Skylab refrigerator test data were not readily available, the model performance was compared with independent analysis. A steady-state condition was achieved within 10 iterations, as seen in the figures. Subsequent changes have been made within the subroutine to allow additional internal iterations, thus achieving steady state conditions within only one or two system iterations. The final total energy input to the locker by conduction and door opening is seen in R(53) and Figure 4-2 to be 6.47 watts (22.06 Btu/hr). This is also equal to the heat transferred to the cooling fluid, as required. For this condition, the coolant fluid temperature rise should be

$$\Delta T_{\text{coolant fluid}} = \frac{q}{\dot{m}\, c_p} = \frac{22.06 \text{ Btu/hr}}{(25 \text{ lb/hr})(0.248 \text{ Btu/lb-°F})} = 3.56\text{°F}$$

For the inlet temperature of 37°F, the outlet coolant should therefore be 40.56°F, which is the temperature shown in Figure 4-1.

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TABLE 4-2

G-189A INPUT DATA FOR REFRIGERATOR CHECKOUT RUN

```
ID**   97    1  REFRIGERATOR - WITH EXTERNAL RADIATOR COOLING CIRCUIT
KBAS   97    3 71                 0   2                                 121
NSTR   97    000010
VARY   97   51 40.                FOOD OUTER SURFACE TEMP (F)
VARY   97   54 70.                AMBIENT GAS TEMP (F)
VARY   97   55 45.                UA, AMBIENT GAS TO INSULATION SURFACE
VARY   97   57 70.                AMBIENT WALL TEMP (F)
VARY   97   58 20.                RADIATION CONDUCTOR, WALL-TO-REFRIGERATOR SURFACE
VARY   97   60 70.                ATTACHED STRUCTURAL TEMP, F
VARY   97   61 .08                CONDUCTOR, STRUCTURE-TO-FOOD OUTER SURFACE
VARY   97   63 70.                INSULATION OUTER SURFACE TEMP, F
VARY   97   64 .7                 THERMAL CONDUCTOR THRU INSULATION

VARY   97   65 2.6                TOTAL INTERNAL VOLUME (CU FT)
VARY   97   69 2.09               PACKAGED FOOD VOLUME (CU FT)
VARY   97   70 1.0                AIR CHANGE PER DOOR OPENING (FRACTION)
VARY   97   71 24.                NUMBER OF DOOR OPENINGS PER DAY
VARY   97   72 26.35              DRY FOOD WEIGHT (LBS)
VARY   97   73 .2                 FRACTION OF FOOD ASSIGNED TO OUTER SURFACE
VARY   97   74 10.                REFRIGERATOR INNER SHELL THERMAL MASS
VARY   97   75 40.                FOOD INNER NODE TEMP (F)
VARY   97   76 4.                 THERMAL CONDUCTIVITY RATIO, FROZEN/UNFROZEN
VARY   97   78 6.3                THERMAL CONDUCTOR, INNER FOOD-TO-OUTER
VARY   97   82 18.                CONDUCTOR, COOLING COILS-TO-FOOD SURFACE
VARY   97   96 70.                AMBIENT AIR TEMP (F)
VARY   97   97 .075               AMBIENT AIR DENSITY (LB/CU FT)
VARY   97   98 0.                 AMBIENT AIR HUMIDITY
VARY   97   99 .24                AMBIENT AIR SPECIFIC HEAT
VARY   97  103 1.                 COOLING COILS THERMAL MASS DRY
VARY   97  104 37.                COOLING COILS TEMP (F)
VARY   97  114 1440.              NEXT TIME (SEC) TO OPEN DOOR
VARY   97  115 .5                 LB H2O/LB OUTER FOOD TOTAL MASS
VARY   97  116 .5                 LB H2O/LB INNER FOOD TOTAL MASS
VARY   97  119 .2                 DRY FOOD SPECIFIC HEAT
```

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TABLE 4-3

FINAL STEADY-STATE SOLUTION FOR REFRIGERATOR CHECKOUT RUN

COMP NO 97 CHILLR SUBR NO 71 PRI SOR 0 SEC SOR 0 COMP PASS NO 10 TIME = .0 SEC

FAILURE FLAGS -- COMP= 0 LOOP= 0

A(1)=	25.000	37.000	.00000	240.00	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPA=	.24800	WTMA=	102.90	RHOA=	86.800	VISCA=	.82200	XKA=	.61200-01
B(1)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPB=	.00000	WTMB=	.00000	RHOB=	.00000	VISCB=	.00000	XKB=	.00000

4-5

D2-118571-1

COMP NO 97 CHILLR SUBR NO 71 PRI SOR 0 SEC SOR 0 COMP PASS NO 10 TIME = .0 SEC

FAILURE FLAGS -- COMP= 0 LOOP= 0 ELAPSED TIME IN MILLISEC -- CP= 392 PP= 0

VR(1)=	25.000	VR(2)=	40.554	VR(3)=	240.00	VR(4)=	240.00	VR(20)=	.00000	VR(21)=	.00000
VR(22)=	.00000	VR(23)=	.00000	VR(51)=	41.779	VR(52)=	.77258	VR(53)=	-22.063	VR(54)=	70.000
VR(55)=	45.000	VR(56)=	-13.460	VR(57)=	70.000	VR(58)=	20.000	VR(59)=	-6.0869	VR(60)=	70.000
VR(61)=	.80000-01	VR(62)=	-2.2578	VR(63)=	69.701	VR(64)=	.70000	VR(65)=	22.063	VR(66)=	40.000
VR(67)=	.00000	VR(68)=	2.6000	VR(69)=	2.0900	VR(70)=	1.0000	VR(71)=	24.000	VR(72)=	26.350
VR(73)=	.20000	VR(74)=	10.000	VR(75)=	41.779	VR(76)=	4.0000	VR(77)=	41.779	VR(78)=	6.3000
VR(79)=	40.554	VR(80)=	.00000	VR(81)=	.00000	VR(82)=	18.000	VR(83)=	.00000	VR(84)=	.00000
VR(85)=	.00000	VR(86)=	.00000	VR(87)=	.00000	VR(88)=	.00000	VR(89)=	.00000	VR(90)=	.00000
VR(91)=	.00000	VR(92)=	.00000	VR(93)=	.00000	VR(94)=	.00000	VR(95)=	.00000	VR(96)=	70.000
VR(97)=	.75000-01	VR(98)=	.00000	VR(99)=	.24000	VR(100)=	.00000	VR(101)=	.00000	VR(102)=	.00000
VR(103)=	1.0000	VR(104)=	40.554	VR(105)=	.25908	VR(106)=	.00000	VR(107)=	.00000	VR(108)=	.00000
VR(109)=	.00000	VR(110)=	.00000	VR(111)=	.00000	VR(112)=	.00000	VR(113)=	.00000	VR(114)=	1440.0
VR(115)=	.50000	VR(116)=	.50000	VR(117)=	.00000	VR(118)=	.00000	VR(119)=	.20000	VR(120)=	.00000
VR(121)=	.00000	VR(122)=	.00000	VR(123)=	.00000	VR(124)=	22.032	VR(125)=	.00000	VR(126)=	.00000
		CPP=	.24800	WTMP=	102.90	RHOP=	86.800	VISCP=	.82200	XKP=	.61200-01
		CPS=	.00000	WTMS=	.00000	RHOS=	.00000	VISCS=	.00000	XKS=	.00000

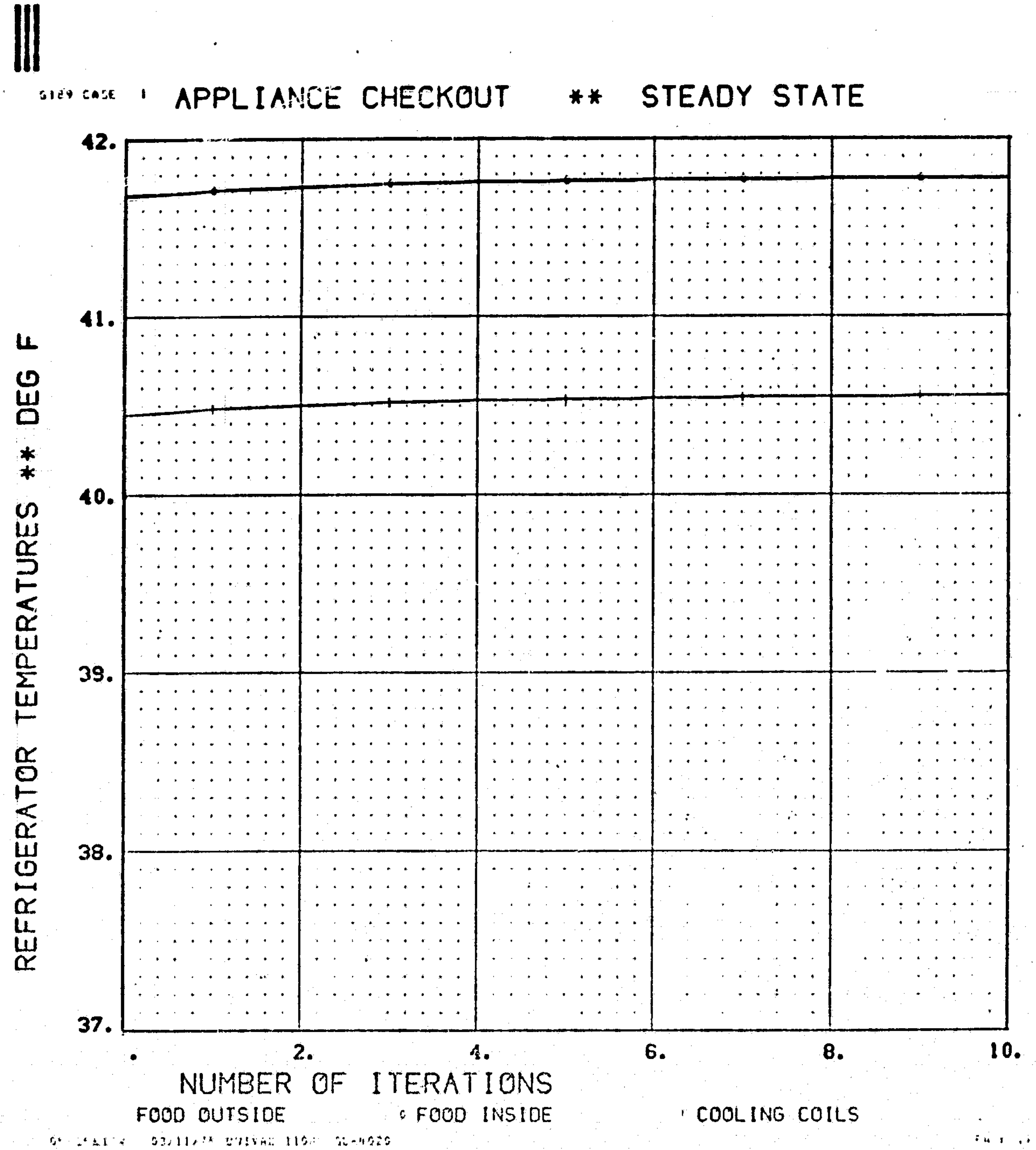

Figure 4-1. Temperatures for Refrigerator Steady-State Checkout Run

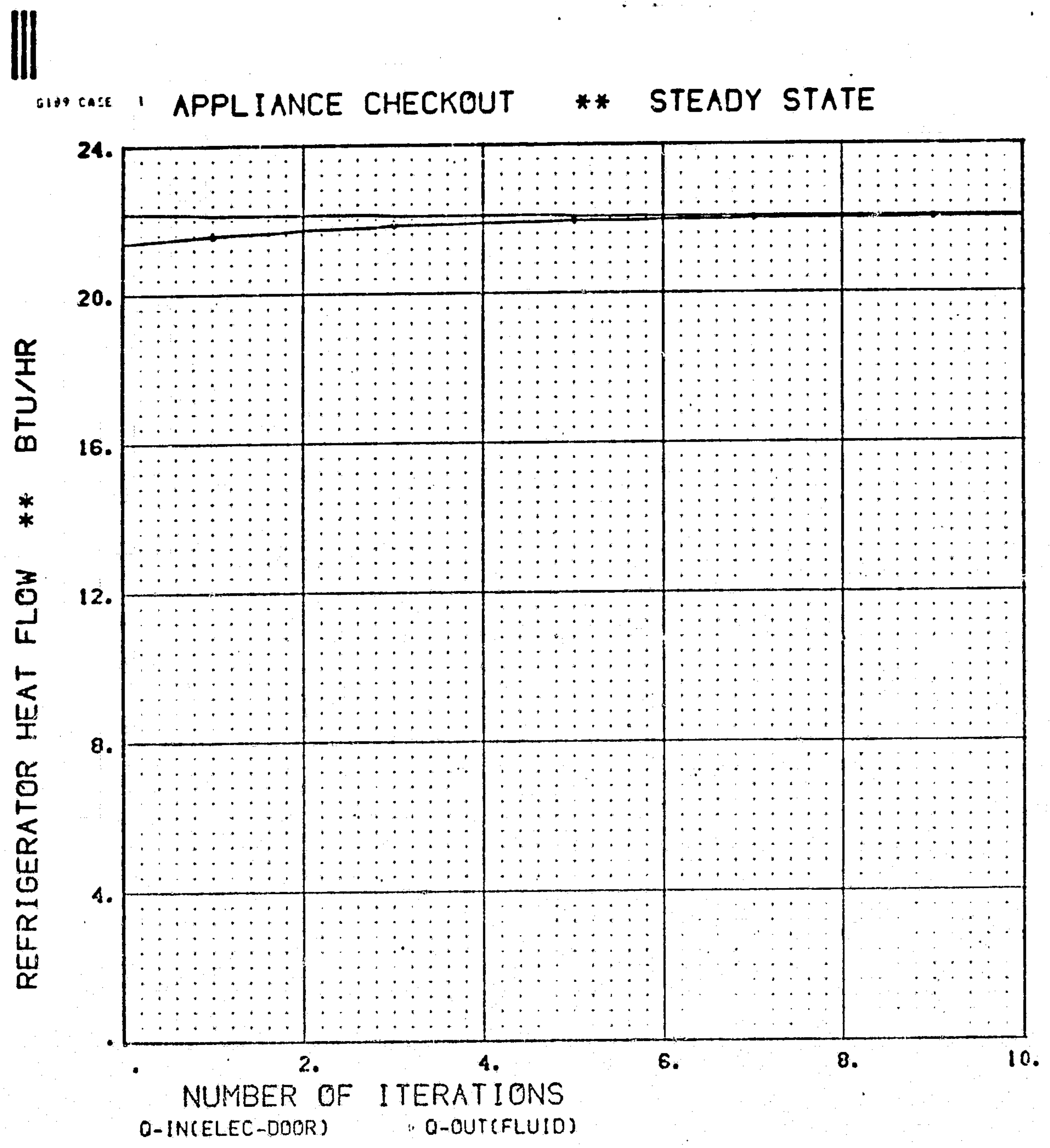

Figure 4-2. Heat Flow for Refrigerator Steady-State Checkout Run

4.1.1 (Continued)

The food inner and outer node temperatures are equal at steady state, as they should be, at 5.43°C (41.78°F). This value results from equating heat in by conduction and door opening to heat out to the fluid. The heat input from door opening `s given by

$$Q_{door} = \rho_{air}\ V\ c_p\ (T_{ambient} - T_{food})$$

$$= 0.075\ \frac{lb}{cu\ ft}\ (0.51\ cu\ ft)\ \left(0.24\ \frac{Btu}{lb°F}\right)\ (70°F\text{-}41.78°F)$$

$$= 0.259\ Btu$$

This agrees with the value of R(105) in Table 4-3. Since 24 door openings per day were assumed in R(71), the steady-state average door opening heat added is 0.076 watts (0.259 Btu/hr). The effective thermal conductance between the food and ambient environment is seen in R(52), Table 4-3 to be 0.773 Btu/hr-°F. This value was determined from Skylab freezer and refrigerator design data. The heat input by conduction to the food is then given by:

$$q_{conduction\ in} = G\ \Delta T$$

$$= 0.773\ \frac{Btu}{hr\ °F}\ (70°F\text{-}41.78°F)$$

$$= 21.80\ Btu/hr$$

Thus, the total heat into the refrigerator by conduction and door opening should be 0.259 + 21.80 = 22.06 Btu/hr = 6.47 watts. As shown previously, this is the same energy input computed by the program; thus, verifying the energy balance made in the subroutine is valid.

4.1.2 CHILLR Transient Refrigerator Case

The G-189A input data for the transient refrigerator run were identical to the steady-state case shown in Table 4-2. A 4-hour transient solution was obtained. The final solution at the end of 4 hours is tabulated in Table 4-4. The transient history is plotted in Figures 4-3 and 4-4. Time did not permit running a longer solution; however, the temperatures in Figure 4-3 are approaching the required steady-state temperatures in Figure 4-1. (Note that a 10-hour refrigerator solution is shown in the Space Station model, Appendix G, for similar input conditions, showing the equilibrium food temperature is indeed approaching the steady-state value in Figure 4-1.)

The heat balance may be verified by checking the energy conservation equation:

$$\Sigma q_{in} - \Sigma q_{out} = \text{heat storage} = m\, c_p \left(\frac{\Delta T}{\Delta t} \right)$$

Checking this equation at a time of 4 hours, the heat input and output are seen in Figure 4-4 to be 22.5 and 17.9 Btu/hr, respectively. The net energy input is therefore 4.6 Btu/hr. The heat storage must be computed as the sum of $(m\, c_p\, \Delta T/\Delta t)$ for the (1) inner food node, (2) outer food/walls/shelves node, and (3) cooling coils. Taking the thermal mass from the solution in Table 4-4 and estimating the heating rates from Figure 4-3 the total heat storage rate at 4 hours time is given by:

$$\text{Heat storage (inner food)} = (25.3\ \text{Btu/°F})(0.123\text{°F/hr})$$

$$= 3.11\ \text{Btu/hr}$$

$$\text{Heat storage (outer food/walls)} = (16.32\ \text{Btu/°F})(0.088\text{°F/hr})$$

$$= 1.43\ \text{Btu/hr}$$

TABLE 4-4

FINAL TRANSIENT SOLUTION FOR REFRIGERATOR CHECKOUT RUN

```
A(1)=   25.000   37.000   .00000   240.00   .00000   .00000   .00000   .00000   .00000   .00000
        .00000   .00000   .00000   .00000   .00000   .00000   .00000   .00000   .00000
           CPA=  .24800      WTMA= 102.90       RHOA= 86.800      VISCA= .82200       XKA= .61200-01
B(1)=   .00000   .00000   .00000   .00000   .00000   .00000   .00000   .00000   .00000   .00000
        .00000   .00000   .00000   .00000   .00000   .00000   .00000   .00000   .00000
           CPB=  .00000      WTMB= .00000       RHOB= .00000      VISCB= .00000       XKB= .00000
```

				VR(97, 65)=	22.495	VR(97, 66)=	40.000	VR(97, 67)=	.00000
VR(97, 68)=	2.6000	VR(97, 69)=	2.0900	VR(97, 70)=	1.0000	VR(97, 71)=	24.000	VR(97, 72)=	26.350
VR(97, 73)=	.20000	VR(97, 74)=	10.000	VR(97, 75)=	40.387	VR(97, 76)=	4.0000	VR(97, 77)=	40.884
VR(97, 78)=	6.3000	VR(97, 79)=	39.886	VR(97, 80)=	.00000	VR(97, 81)=	.00000	VR(97, 82)=	18.000
VR(97, 83)=	.00000	VR(97, 84)=	.00000	VR(97, 85)=	.00000	VR(97, 86)=	.00000	VR(97, 87)=	.00000
VR(97, 88)=	.00000	VR(97, 89)=	.00000	VR(97, 90)=	.00000	VR(97, 91)=	.00000	VR(97, 92)=	.00000
VR(97, 93)=	.00000	VR(97, 94)=	16.324	VR(97, 95)=	25.296	VR(97, 96)=	70.000	VR(97, 97)=	.75000-01
VR(97, 98)=	.00000	VR(97, 99)=	.24000	VR(97,100)=	.00000	VR(97,101)=	.00000	VR(97,102)=	.00000
VR(97,103)=	1.0000	VR(97,104)=	39.886	VR(97,105)=	.00000	VR(97,106)=	.00000	VR(97,107)=	.00000
VR(97,108)=	.00000	VR(97,109)=	.00000	VR(97,110)=	.00000	VR(97,111)=	.00000	VR(97,112)=	.00000
VR(97,113)=	.00000	VR(97,114)=	15840.	VR(97,115)=	.50000	VR(97,116)=	.50000	VR(97,117)=	.00000
VR(97,118)=	.00000	VR(97,119)=	.20000	VR(97,120)=	.00000	VR(97,121)=	.00000	VR(97,122)=	.00000
VR(97,123)=	.00000	VR(97,124)=	17.896	VR(97,125)=	.00000	VR(97,126)=	.00000		

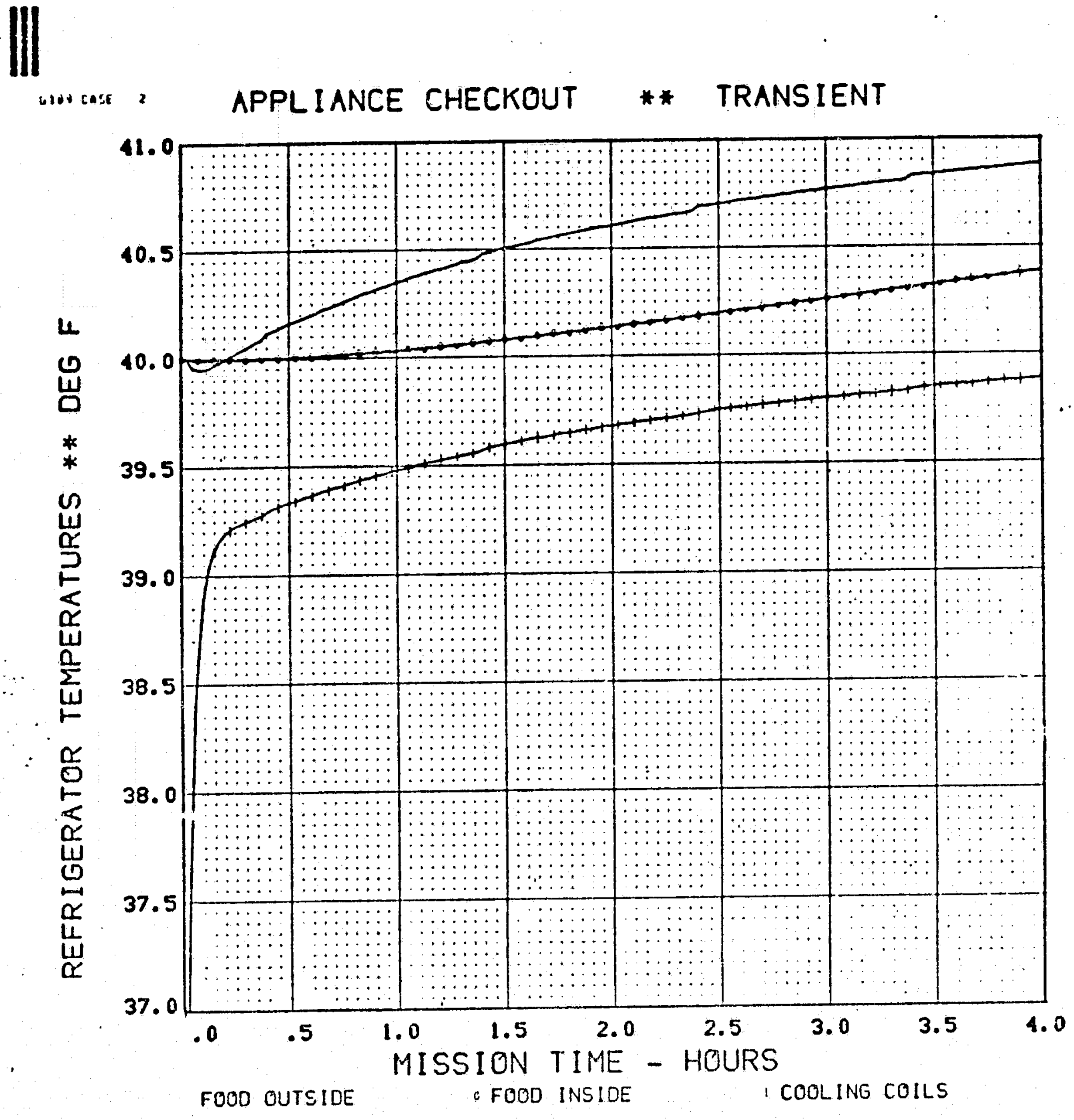

Figure 4-3. Temperatures for Refrigerator Transient Checkout Run

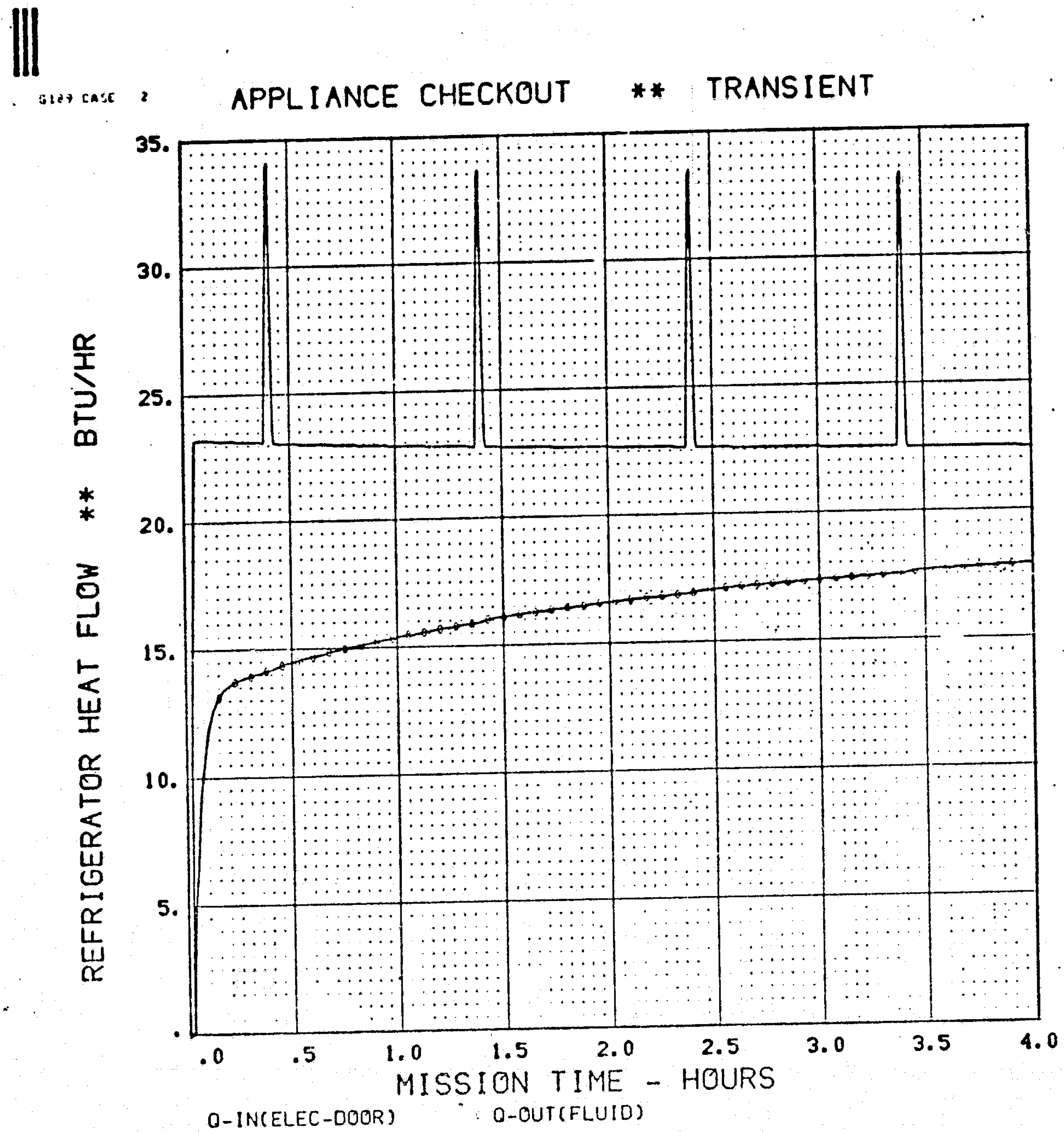

Figure 4-4. Heat Flow for Refrigerator Transient Checkout Run

4.1.2 (Continued)

$$\text{Heat storage (cooling coils)} = (1.0\ \text{Btu/°F})(0.063\text{°F/hr})$$
$$= 0.06\ \text{Btu/hr}$$

Thus, the combined heat storage rate at 4 hours time is 4.60 Btu/hr. As shown previously, this is also equal to the net heat input to the refrigerator, thus verifying the accuracy of the overall transient energy balance.

4.1.3 <u>CHILLR Steady-State Freezer Case</u> (Vapor Compression)

The G-189A component input data for the freezer case are listed in Table 4-5. The basic freezer locker design is identical to the refrigerator locker discussed previously, except that the cooling fluid is supplied by a self-contained refrigeration unit with a coefficient of performance of 1.07. For this steady-state case, the refrigeration unit was assumed to be on continuously at normal operating capacity. The final steady-state solution is tabulated in Table 4-6. The results for the run are plotted in Figures 4-5 through 4-7.

The net heat conducted into the food locker from the ambient environment is seen in Figure 4-6 to be 74.9 Btu/hr. At steady state, all this heat must be transferred directly to the cooling coils. The coefficient of

TABLE 4-5

G-189A INPUT DATA FOR FREEZER CHECKOUT RUN

ID**	62	1		FREEZER - WITH SELF CONTAINED COOLING UNIT
KBAS	62	0 71 7		2 97
NSTR	62	0101111		
KARY	62	17	0	FREEZER INITIAL CHILLER OPERATION
VARY	62	51	-10.	FOOD OUTER SURFACE TEMP (F)
VARY	62	54	70.	AMBIENT GAS TEMP (F)
VARY	62	55	45.	UA, AMBIENT GAS-TO-INSULATION SURFACE
VARY	62	57	70.	AMBIENT WALL TEMP (F)
VARY	62	58	20.	RADIATION CONDUCTOR, WALL-TO-FREEZER SURFACE
VARY	62	60	70.	ATTACHED STRUCTURAL TEMP (F)
VARY	62	61	.08	CONDUCTOR, STRUCTURE-TO-FOOD OUTER SURFACE
VARY	62	63	70.	INSULATION OUTER SURFACE TEMP (F)
VARY	62	64	.7	THERMAL CONDUCTOR THRU INSULATION
VARY	62	66	-10.	CONTROL TEMP (F)
VARY	62	67	1.0	DUTY CYCLE
VARY	62	68	2.6	TOTAL INTERNAL VOLUME (CU FT)
VARY	62	69	2.09	PACKAGED FOOD VOLUME (CU FT)
VARY	62	70	1.0	AIR CHANGE PER DOOR OPENING (FRACTION)
VARY	62	72	26.35	DRY FOOD WEIGHT (LBS)
VARY	62	73	.2	FRACTION OF FOOD ASSIGNED TO OUTER SURFACE
VARY	62	74	10.	FREEZER INNER SHELL THERMAL MASS
VARY	62	75	-10.	FOOD INNER NODE TEMP (F)
VARY	62	76	4.	THERMAL CONDUCTIVITY RATIO, FROZEN/UNFROZEN
VARY	62	77	-10.	AIR TEMP INSIDE FOOD COMPARTMENT
VARY	62	78	6.3	THERMAL CONDUCTOR, UNFROZEN FOOD INNER-TO-OUTER
VARY	62	79	-10.	AIR TEMP IN CHILLING COMPARTMENT (F)
VARY	62	82	6.	CONDUCTOR, COOLING COILS-TO-FOOD SURFACE
VARY	62	83	-15.	LOW TEMP LIMIT TO TURN CHILLER OFF (F)
VARY	62	84	-5.	HIGH TEMP LIMIT TO TURN CHILLER ON (F)
VARY	62	96	70.	AMBIENT AIR TEMP (F)
VARY	62	97	.075	AMBIENT AIR DENSITY (LB/CU FT)
VARY	62	98	0.	AMBIENT AIR HUMIDITY
VARY	62	99	.24	AMBIENT AIR SPECIFIC HEAT
VARY	62	100	1.07	CHILLER UNIT COP
VARY	62	102	-28.	EVAPORATOR DESIGN TEMP (F)
VARY	62	103	1.	COOLING COILS THERMAL MASS DRY
VARY	62	104	-10.	COOLING COILS TEMP (F)
VARY	62	115	.5	LB H2O/LB OUTER FOOD TOTAL MASS
VARY	62	116	.5	LB H2O/LB INNER FOOD TOTAL MASS
VARY	62	119	.2	DRY FOOD SPECIFIC HEAT
VARY	62	120	.08	CONDUCTOR,COOLING COILS-TO-CHILLER UNIT (OFF)
VARY	62	121	70.	CHILLER UNIT TEMP (F) ATTACHED TO COOLING COILS
VARY	62	125	.25	FIRST DOOR OPENING TIME (HRS)
VARY	62	126	.5	SECOND DOOR OPENING TIME (HRS)
VARY	62	127	.75	THIRD DOOR OPENING TIME (HRS)
VARY	62	128	1.2	FOURTH DOOR OPENING TIME (HRS)
VARY	62	129	1.7	FIFTH DOOR OPENING TIME (HRS)
VARY	62	130	2.	SIXTH DOOR OPENING TIME (HRS)

TABLE 4-6

FINAL STEADY-STATE SOLUTION FOR FREEZER CHECKOUT RUN

COMP NO 62 CHILLR SUBR NO 71 PRI SOR 0 SEC SOR 0 COMP PASS NO 10 TIME = .0 SEC
FAILURE FLAGS -- COMP= 0 LOOP= 0

A(1)=	64.000	70.000	.00000	14.700	63.501	.49920	.00000	.24122	28.902	14.720
	48.394	.38720	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPA=	.24277	WTMA=	28.766	RHOA=	.74362-01	VISCA=	.47000-01	XKA=	.15400-01
B(1)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPB=	.00000	WTMB=	.00000	RHOB=	.00000	VISCB=	.00000	XKB=	.00000

VR(1)=	64.000	VR(2)=	79.328	VR(3)=	14.700	VR(4)=	14.700	VR(5)=	63.501	VR(6)=	.49920
VR(7)=	.00000	VR(8)=	.24122	VR(9)=	28.902	VR(10)=	14.720	VR(11)=	48.394	VR(12)=	.38720
VR(13)=	.00000	VR(14)=	.00000	VR(20)=	.00000	VR(21)=	.00000	VR(22)=	.00000	VR(23)=	.00000
VR(51)=	-16.820	VR(52)=	.77258	VR(53)=	-74.9	VR(54)=	70.000	VR(55)=	45.000	VR(56)=	-41.430
VR(57)=	70.000	VR(58)=	20.000	VR(59)=	-18.703	VR(60)=	70.000	VR(61)=	.80000-01	VR(62)=	-6.9456
VR(63)=	69.079	VR(64)=	.70000	VR(65)=	74.9	VR(66)=	-10.000	VR(67)=	1.0	VR(68)=	2.6000
VR(69)=	2.0900	VR(70)=	1.0000	VR(71)=	.00000	VR(72)=	26.350	VR(73)=	.20000	VR(74)=	10.000
VR(75)=	-16.820	VR(76)=	4.0000	VR(77)=	-16.820	VR(78)=	6.3000	VR(79)=	-26.000	VR(80)=	.00000
VR(81)=	.00000	VR(82)=	6.0000	VR(83)=	-15.000	VR(84)=	-5.0000	VR(85)=	.00000	VR(86)=	.00000
VR(87)=	.00000	VR(88)=	.00000	VR(89)=	.00000	VR(90)=	.00000	VR(91)=	.00000	VR(92)=	.00000
VR(93)=	.00000	VR(94)=	.00000	VR(95)=	.00000	VR(96)=	70.000	VR(97)=	.75000-01	VR(98)=	.00000
VR(99)=	.24000	VR(100)=	1.0700	VR(101)=	.00000	VR(102)=	-28.000	VR(103)=	1.0000	VR(104)=	-28.000
VR(105)=	.00000	VR(106)=	.00000	VR(107)=	.00000	VR(108)=	.00000	VR(109)=	20.515	VR(110)=	.00000
VR(111)=	.00000	VR(112)=	.00000	VR(113)=	.00000	VR(114)=	.00000	VR(115)=	.50000	VR(116)=	.50000
VR(117)=	.00000	VR(118)=	.00000	VR(119)=	.20000	VR(120)=	.80000-01	VR(121)=	70.000	VR(122)=	.00000
VR(123)=	70.000	VR(124)=	144.94	VR(125)=	.25000	VR(126)=	.50000	VR(127)=	.75000	VR(128)=	1.2000
VR(129)=	1.7000	VR(130)=	2.0000	VR(131)=	.00000	VR(132)=	67.079	VR(133)=	.00000	VR(	
		CPP=	.24277	WTMP=	28.766	RHOP=	.73076-01	VISCP=	.47000-01	XKP=	.15400-01
		CPS=	.00000	WTMS=	.00000	RHOS=	.00000	VISCS=	.00000	XKS=	.00000

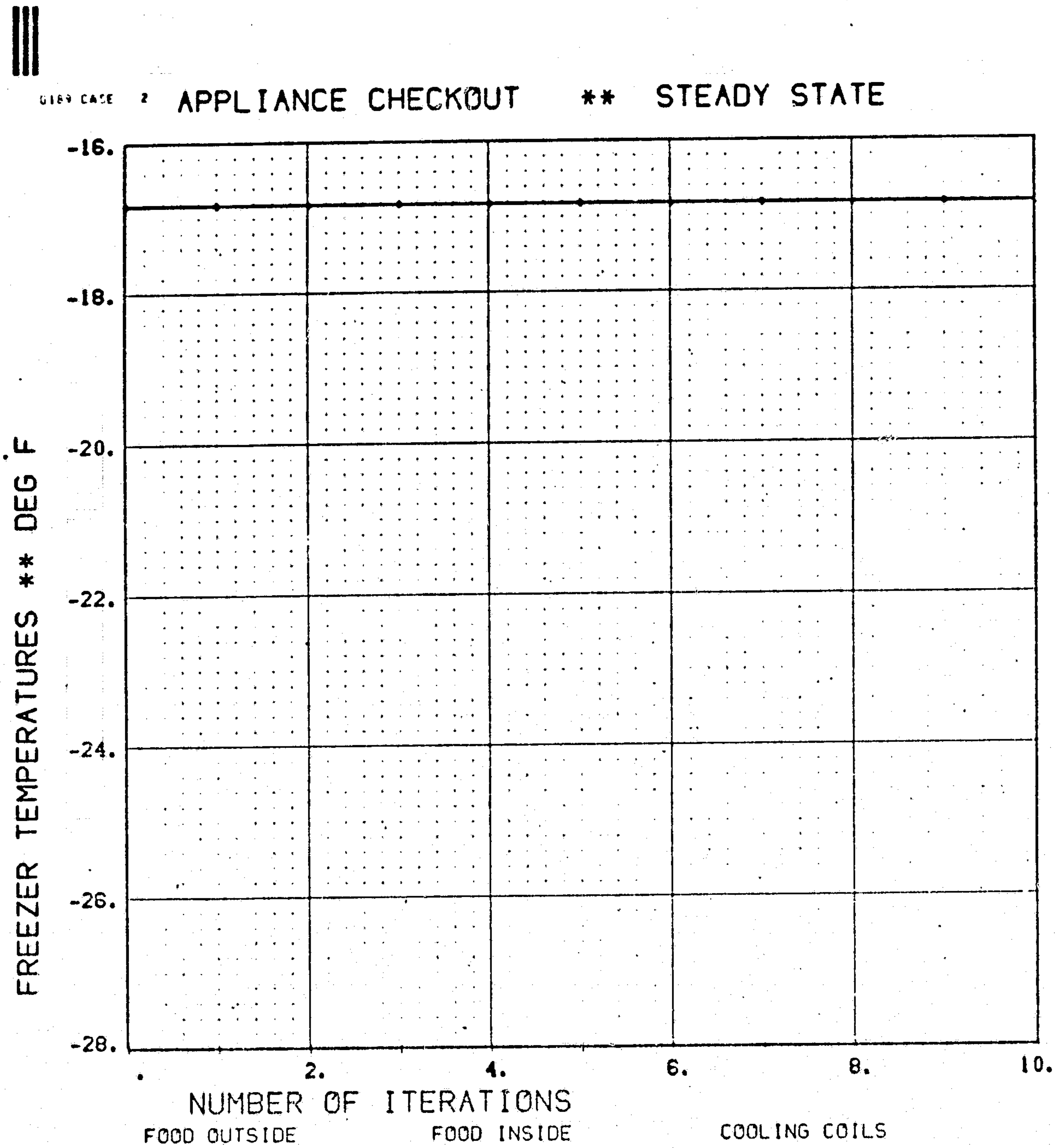

Figure 4-5. Temperatures for Freezer Steady-State Checkout Run

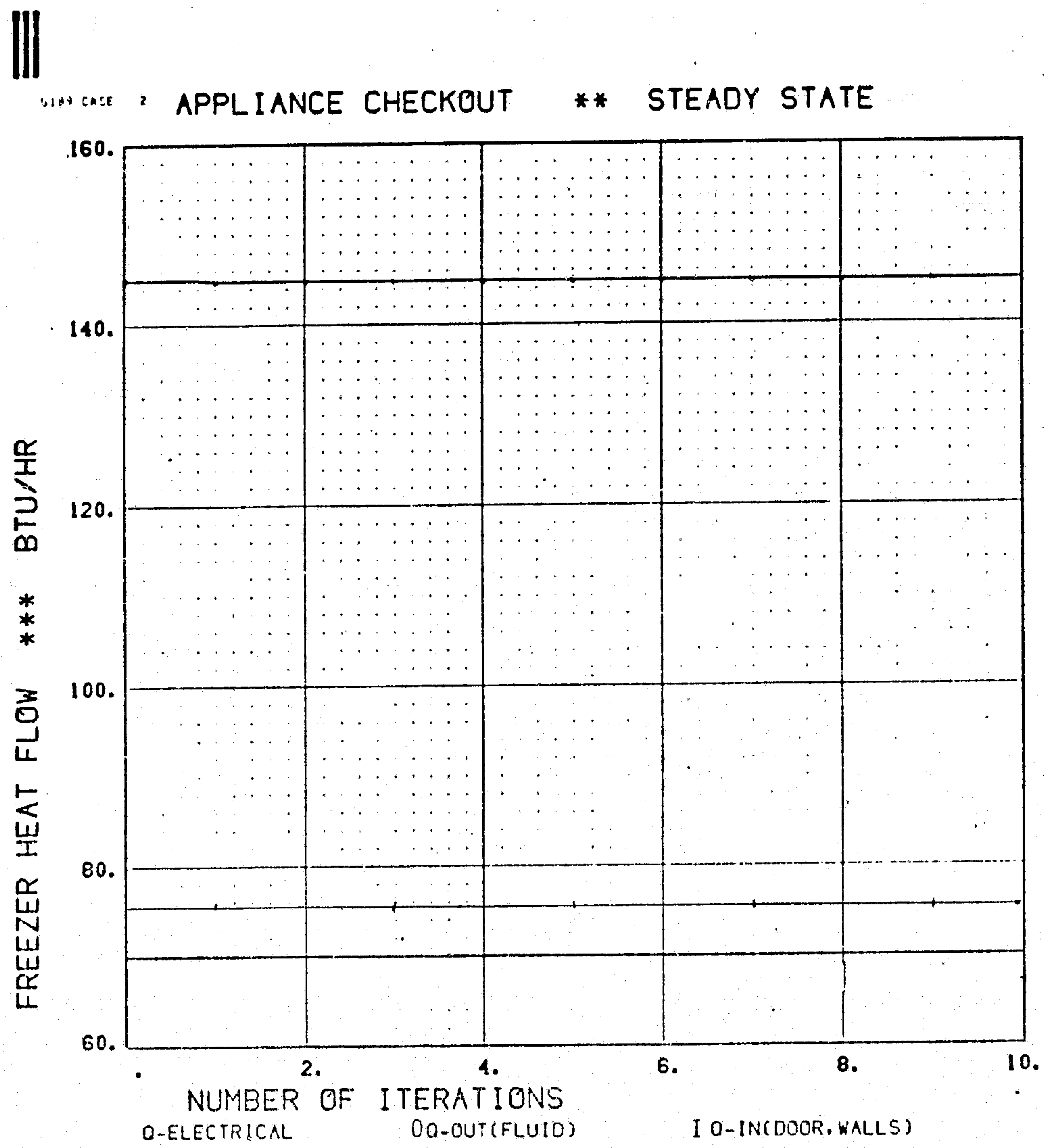

Figure 4-6. Heat Flow for Freezer Steady-State Checkout Run

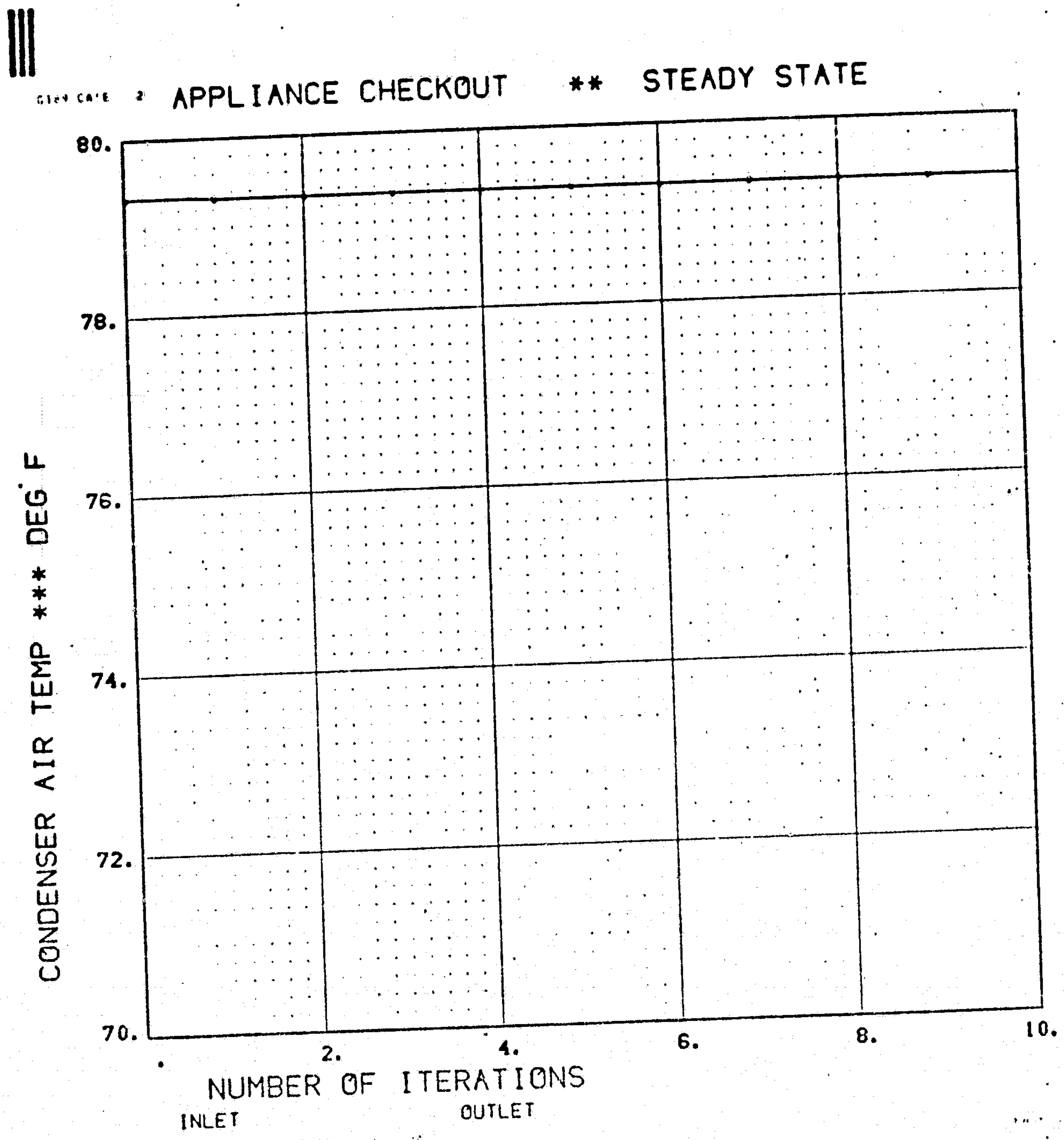

Figure 4-7. Condenser Air Temperatures for Freezer Steady-State Checkout Run

4.1.3 (Continued)

performance for the cooling unit was input in R(100) as 1.07 according to Reference 22. Thus, from equation (3.1.15), the total electrical input power required should be

$$q_E = \frac{q_L}{COP} = \frac{74.9}{1.07} = 70.0 \text{ Btu/hr}$$

This value is correctly reflected in Figure 4-6. From equation (3.1.16), the total heat rejected by the cooling unit condenser to ambient should be

$$q_H = q_L + q_E = 74.9 + 70.0 = 144.9 \text{ Btu/hr}$$

This value is also correctly reflected in Figure 4-6, thus verifying the accuracy of the thermal energy balance.

4.1.4 CHILLR Transient Freezer Case

The G-189A input data for the transient freezer run were identical to the steady-state case shown in Table 4-5. The basic freezer locker design is identical to the refrigerator locker discussed in Paragraphs 4.1.1 and 4.1.2, except that the cooling fluid is supplied by a self-contained refrigeration unit with a coefficient of performance of 1.07. A 4-hour transient solution was obtained. Initially, the refrigeration unit was turned off. The locker gradually warmed until the food temperature reached -5°F (after about 2 hours), when the refrigeration unit was turned on. The cooling unit remained on for the remainder of the run. The final results at the end of 4 hours are tabulated in Table 4-7. The transient history is plotted in Figures 4-8 through 4-10. The solution during the second 2 hours, when the cooling unit is turned on, is approaching the steady-state case discussed in Paragraph 4.1.3 and plotted in Figures 4-5 through 4-7. The temperatures and heat balance were checked against the analytical model in the same manner as was done for the previous cases and found to be valid.

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TABLE 4-7

FINAL TRANSIENT SOLUTION FOR FREEZER CHECKOUT RUN

COMP NO 62 CHILLR SUBR NO 71 PRI SOR 0 SEC SOR 0 COMP PASS NO 153 TIME = 13770.0 SEC
FAILURE FLAGS -- COMP= 0 LOOP= 0

A(1)=	64.000	70.000	.00000	14.700	63.501	.49920	.00000	.24122	28.902	14.720
	48.394	.38720	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPA=	.24277	WTMA=	28.766	RHOA=	.74362-01	VISCA=	.47000-01	XKA=	-.15400-01
B(1)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPB=	.00000	WTMB=	.00000	RHOB=	.00000	VISCB=	.00000	XKB=	.00000

R(1)=	64.000	84.285	14.700	14.700	63.501	.49920	.00000	.24122	28.902	14.720
R(11)=	48.394	.38720	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(21)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(31)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(41)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(51)=	-10.186	.77258	-68.812	70.000	45.000	-38.243	70.000	20.000	-17.268	70.000
R(61)=	.80000-01	-6.4118	69.150	.70000	113.42	-10.000	1.0000	2.6000	2.0900	1.0000
R(71)=	.00000	26.350	.20000	10.000	-9.2204	4.0000	-10.186	6.3000	-28.000	.00000
R(81)=	.00000	6.0000	-15.000	-5.0000	.00000	.00000	.00000	.00000	.00000	.00000
R(91)=	.00000	.00000	.00000	13.478	13.913	70.000	.75000-01	.00000	.24000	1.0700
R(101)=	.00000	-28.000	1.0000	-28.000	.00000	.00000	.00000	.00000	31.067	.00000
R(111)=	.00000	.00000	.00000	.00000	.50000	.50000	1.0000	1.0000	.20000	.60000-01
R(121)=	70.000	.00000	70.000	221.95	24.250	24.500	24.750	25.200	25.700	26.000
R(131)=	.00000	61.923	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000

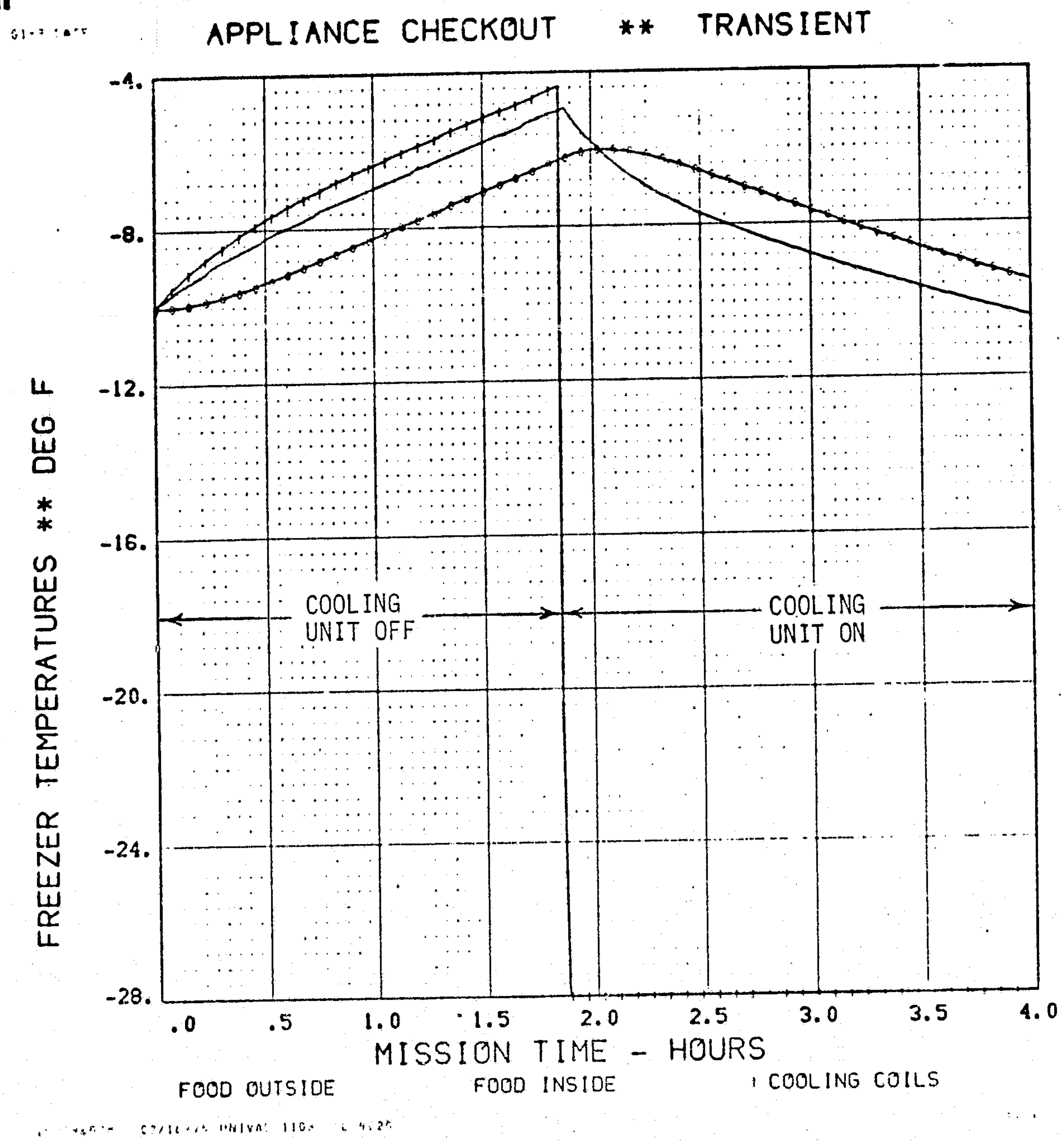

Figure 4-8. Temperatures for Freezer Transient Checkout Run

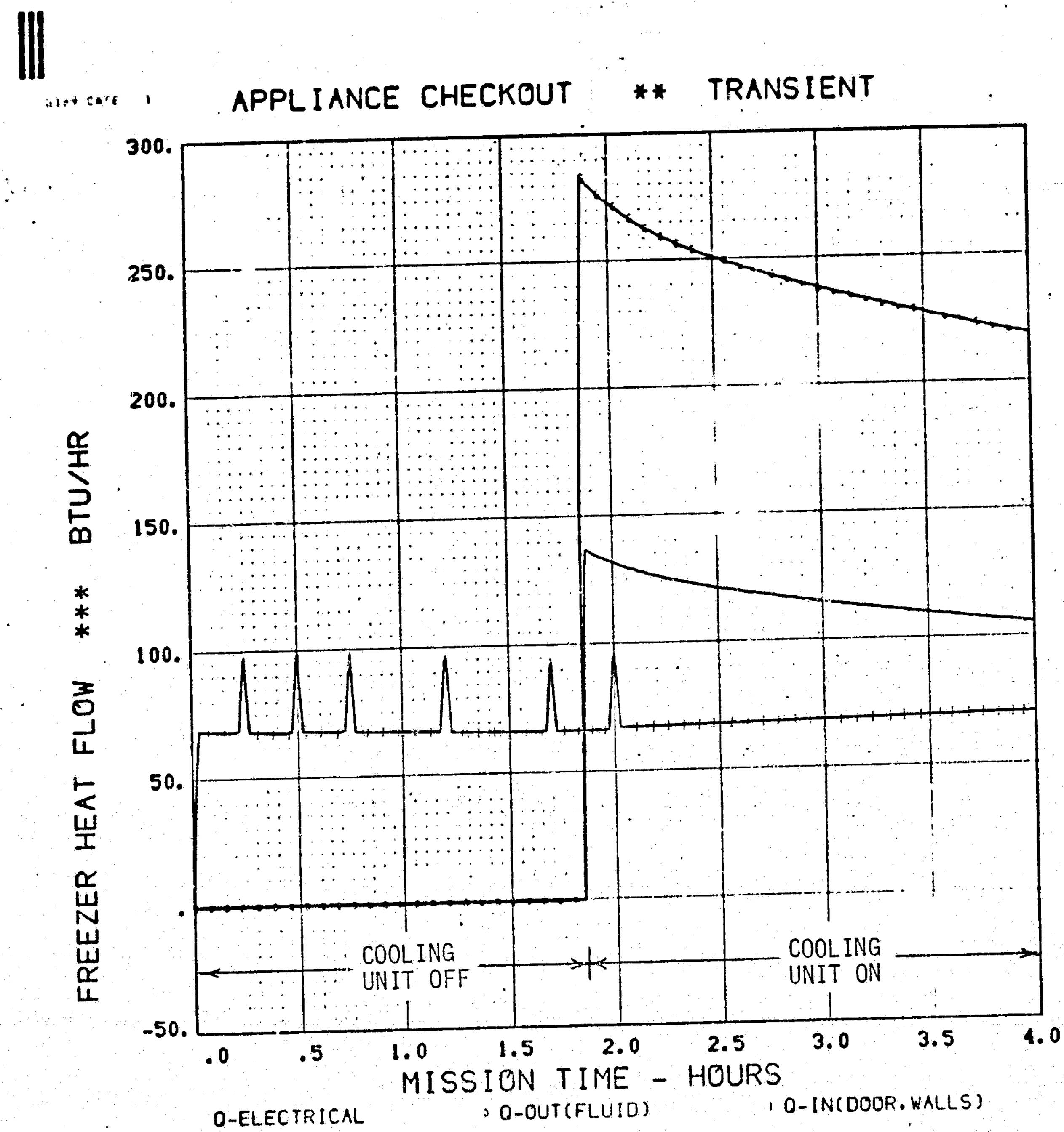

Figure 4-9. Heat Flow for Freezer Transient Checkout Run

C-3

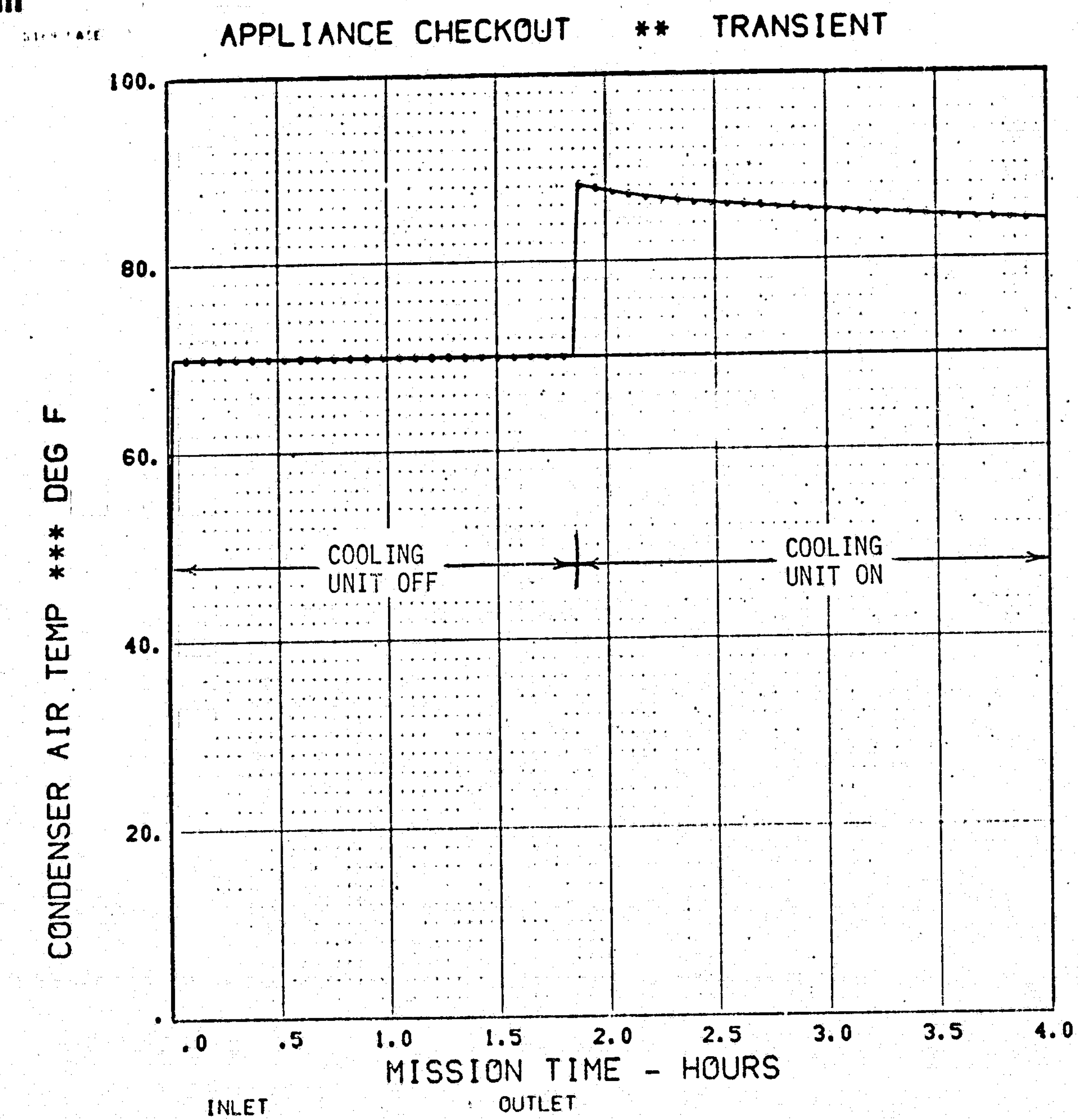

Figure 4-10. Condenser Air Temperatures for Freezer Transient Checkout Run

4.1.5 CHILLR Simulation of Shuttle Stirling Cycle Freezer

A preliminary design of a food and medical sample freezer using a Stirling cycle refrigeration unit was developed for use on Shuttle (Reference 32). The CHILLR subroutine has been used to simulate that freezer, and excellent agreement has been obtained. The model input data for this simulation have been built into the subroutine as default data. These data are listed in Table 3-2.

A detailed three-dimensional thermal model of the freezer is described in Reference 32. In the absence of hardware test data, the analytical model results were used to correlate the G-189A subroutine model. Both the G-189A subroutine and the independent three-dimensional model described above were run with identical operating conditions, and their results compared in Figures 4-11 through 4-16. The transient run, Figures 4-14 through 4-16, included five door openings near time 3 hours, and a fresh medical sample (with 2.15 lbs. water) initially at 80°F inserted at 6 hours. The results from the two models agree very closely--especially the overall heat balance, which is the most important G-189A consideration. For steady state operating conditions in a 70°F ambient environment with no door openings or medical sample insertion, the refrigeration unit duty cycle (fraction of total time on) for both models was 62 percent.

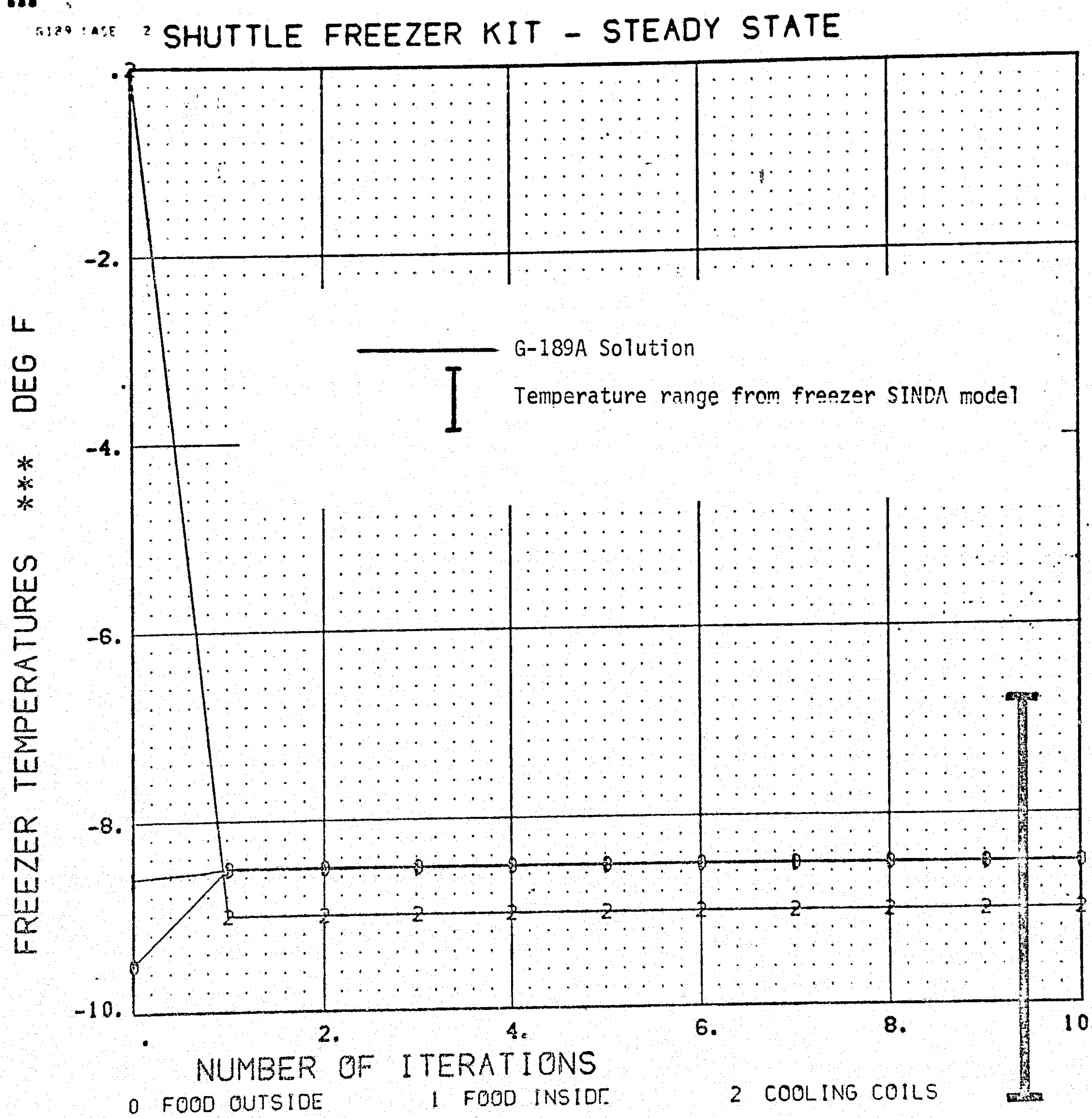

Figure 4-11. Correlation of G-189A Shuttle Freezer Steady State Temperatures

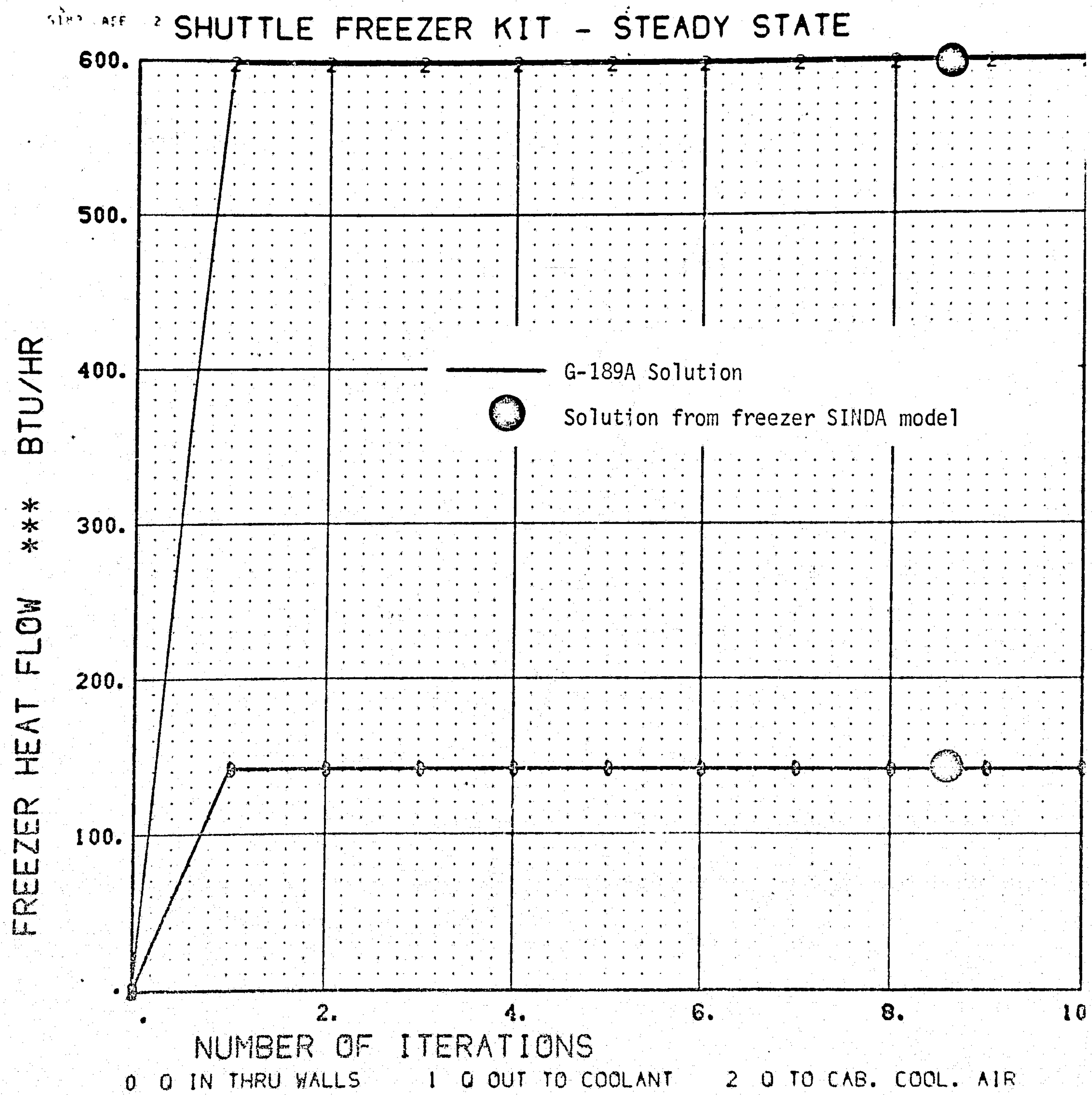

Figure 4-12. Correlation of G-189A Shuttle Freezer Steady State Thermal Solution

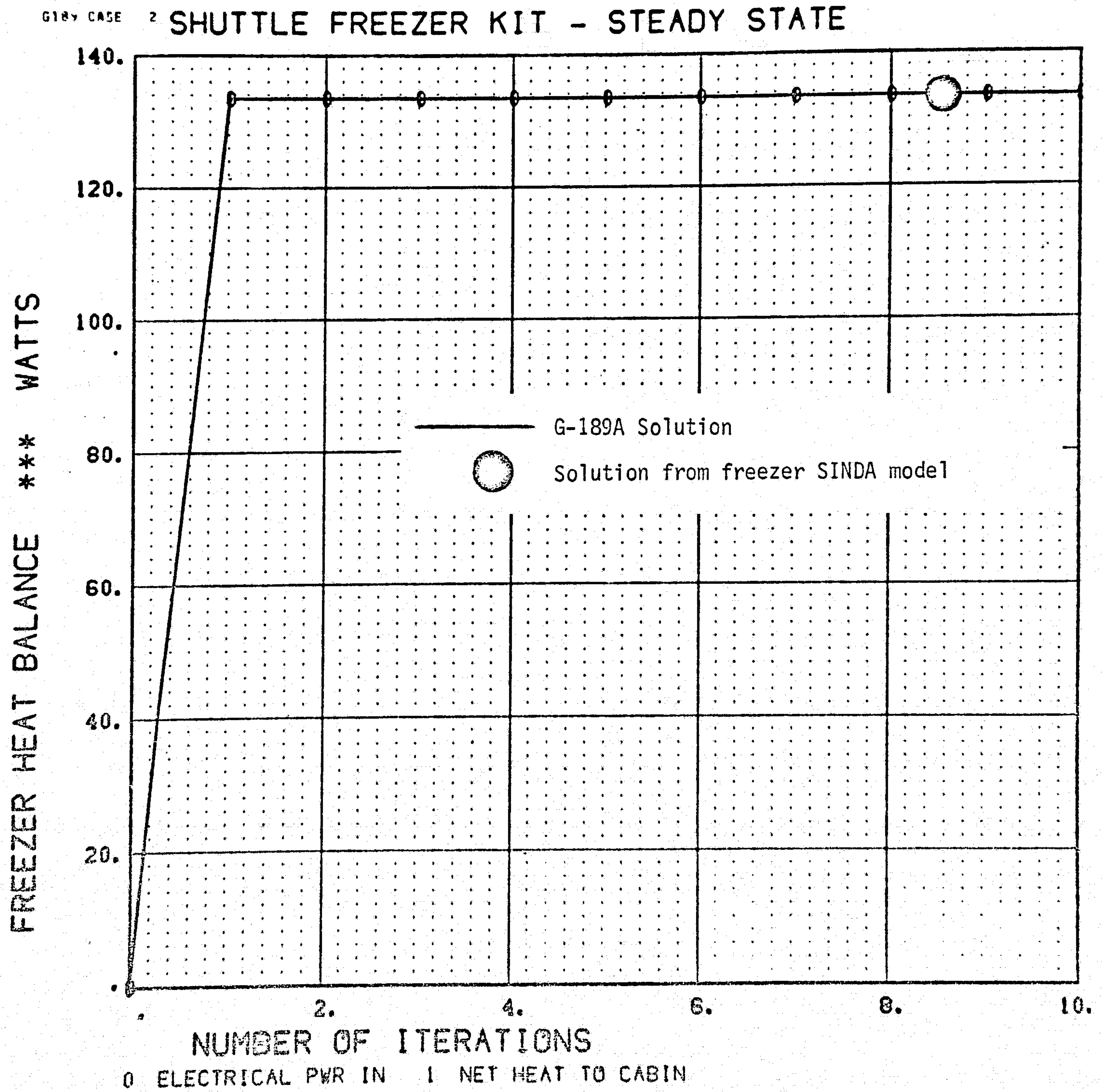

Figure 4-13. Correlation of G-189A Shuttle Freezer Steady State Heat Balance

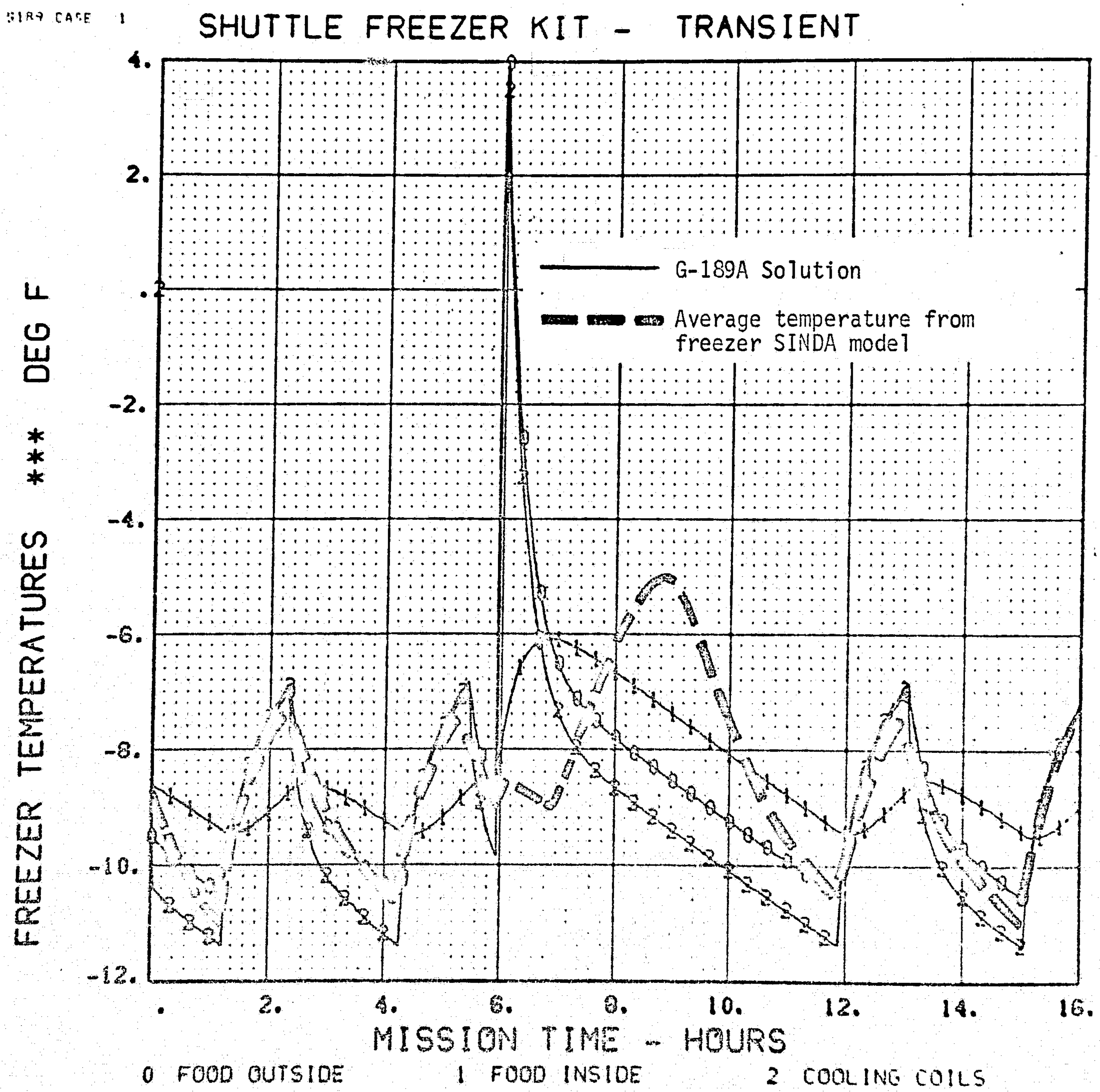

Figure 4-14. Correlation of G-189A Shuttle Freezer Transient Temperatures

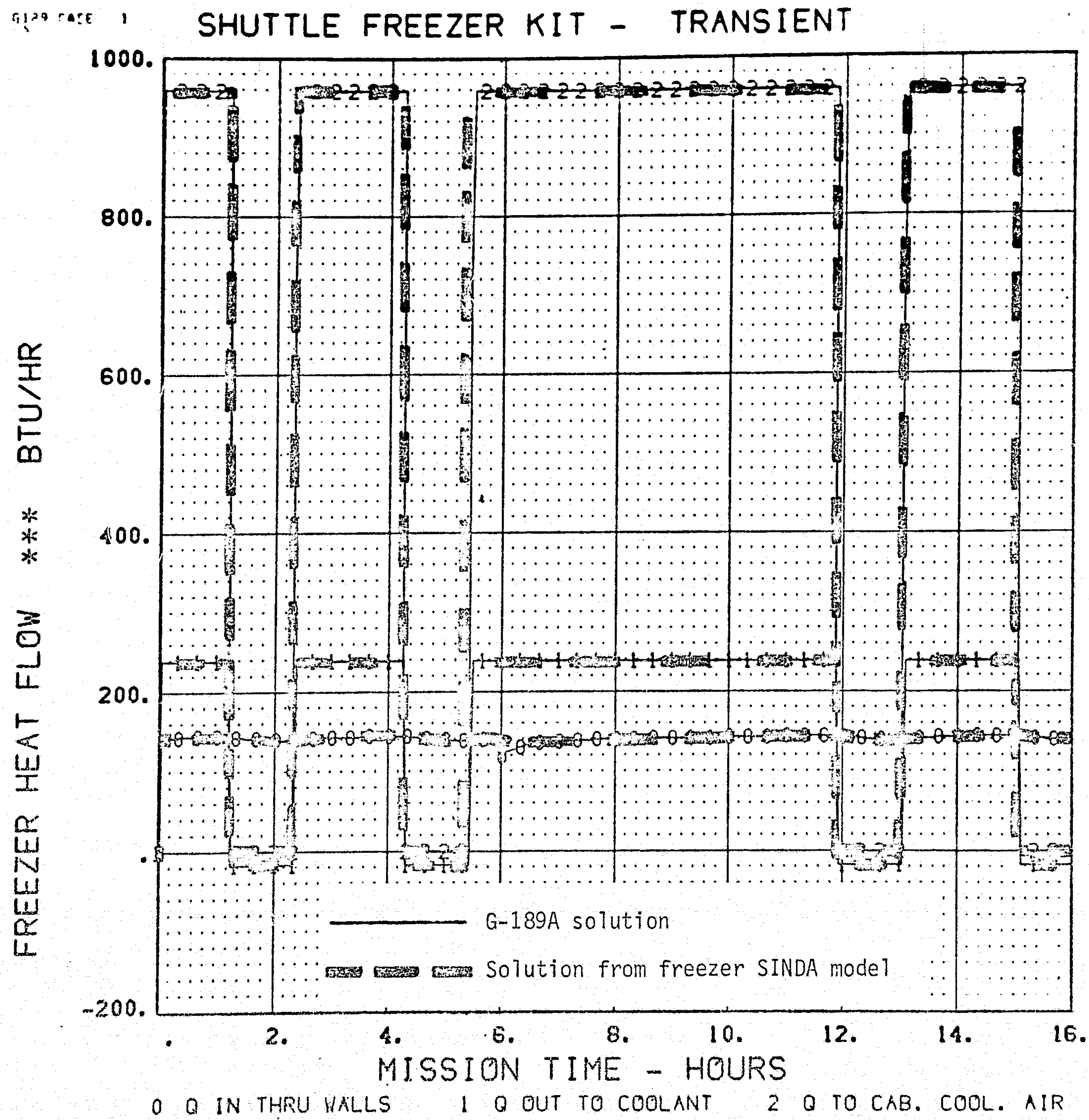

Figure 4-15. Correlation of G-189A Shuttle
Freezer Transient Thermal Solution

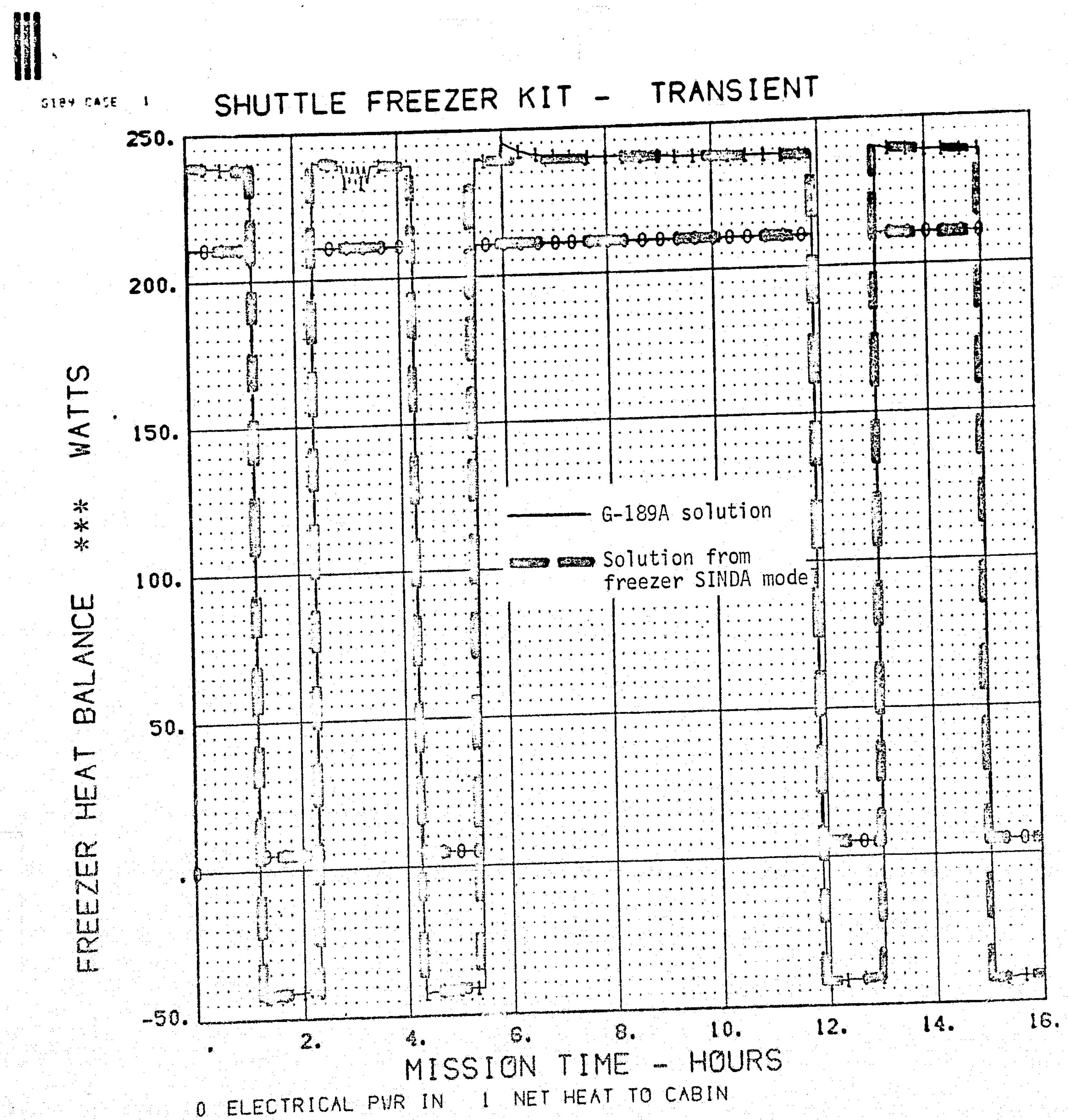

Figure 4-16. Correlation of G-189A Shuttle Freezer Transient Heat Balance

4.2 FTRAY CHECKOUT RUN

The FTRAY subroutine was used to model a standard Skylab-type food heating tray. Two cases were run: one steady state and one transient. For both cases the subroutine input default data, listed in Table 3-5, were used to simulate heating of a frozen meat product stored at -25°F to approximately 150°F.

4.2.1 FTRAY Steady-State Case

The final component data for the steady-state food tray case are listed in Table 4-8. The results for the run are plotted in Figures 4-17 and 4-18. The temperature data are compared with actual test data from Reference 23, both for an average heater power of 7.9 watts. The test data were taken with a single thermocouple buried roughly near the center of the heated food. Since such a large temperature gradient exists within the food (43°F), it is not possible to compare temperatures at specific nodes with the test data point. However, the test data fall generally within the computed temperature range. In Figure 4-18, the heat dissipated to ambient surroundings is equal to the input heater power of 7.9 watts (26.95 Btu/hr) at steady state, as required by an energy balance on the component.

4.2.2 FTRAY Transient Case

The final component data for the transient food tray run are listed in Table 4-9. The results for 3 hours of heating time are plotted in Figures 4-19 and 4-20. This time was long enough to reach steady-state conditions, and the final temperatures are seen to reach the same values shown in Figure 4-17 for the steady-state case. Transient test data from Reference 23 are plotted in Figure 4-19. These data were taken with a single thermocouple without a clearly defined location, but which was roughly near the center of the food. Since the temperature gradients within the food are so large, the test temperatures cannot be compared with any specific food node. However, they do fall roughly within the computed temperature

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TABLE 4-8

FINAL STEADY-STATE SOLUTION FOR FOOD TRAY CHECKOUT RUN

COMP NO 2 FTRAY SUBR NO 66 PRI SOR 0 SEC SOR 0 COMP PASS NO 10 TIME = .0 SEC

FAILURE FLAGS -- COMP= 0 LOOP= 0

A(1)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPA=	.00000	WTMA=	.00000	RHOA=	.00000	VISCA=	.00000	XKA=	.0C000
B(1)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00006	.00000	
	CP8=	.00000	WTMB=	.00000	RHOB=	.00000	VISCB=	.00000	XKB=	.00000

VR(1)=	.00000	VR(2)=	.00000	VR(3)=	.00000	VR(4)=	.00000	VR(20)=	.00000	VR(21)=	.00000
VR(22)=	.00000	VR(23)=	.00000	VR(51)=	136.94	VR(52)=	.43418	VR(53)=	26.955	VR(54)=	70.000
VR(55)=	.14800	VR(56)=	8.9978	VR(57)=	70.000	VR(58)=	.62500-01	VR(59)=	4.7299	VR(60)=	70.000
VR(61)=	.21600	VR(62)=	13.159	VR(63)=	136.93	VR(64)=	.00000	VR(65)=	26.955	VR(66)=	1.0000
VR(67)=	1.8750	VR(68)=	1.1250	VR(69)=	7.9000	VR(70)=	.62489	VR(71)=	158.54	VR(72)=	137.68
VR(73)=	115.61	VR(74)=	126.46	VR(75)=	146.42	VR(76)=	.11379	VR(77)=	.25566	VR(78)=	.22361
VR(79)=	.25544	VR(80)=	.21818	VR(81)=	.56364	VR(82)=	.99000	VR(83)=	.25000	VR(84)=	.00000
VR(85)=	2.0000	VR(86)=	.00000	VR(87)=	.00000	VR(88)=	.00000	VR(89)=	.00000	VR(90)=	.00000
VR(91)=	.00000	VR(92)=	.00000	VR(93)=	.00000	VR(94)=	.00000	VR(95)=	.00000	VR(96)=	.00000
VR(97)=	.00000	VR(98)=	.00000	VR(99)=	.00000	VR(100)=	.00000	VR(101)=	.00000	VR(102)=	.00000
VR(103)=	.00000	VR(104)=	1.0000	VR(105)=	.00000	VR(					
		CPP=	.00000	WTMP=	.00000	RHOP=	.00000	VISCP=	.00000	XKP=	.00000
		CPS=	.00000	WTMS=	.00000	RHOS=	.00000	VISCS=	.00000	XKS=	.00000

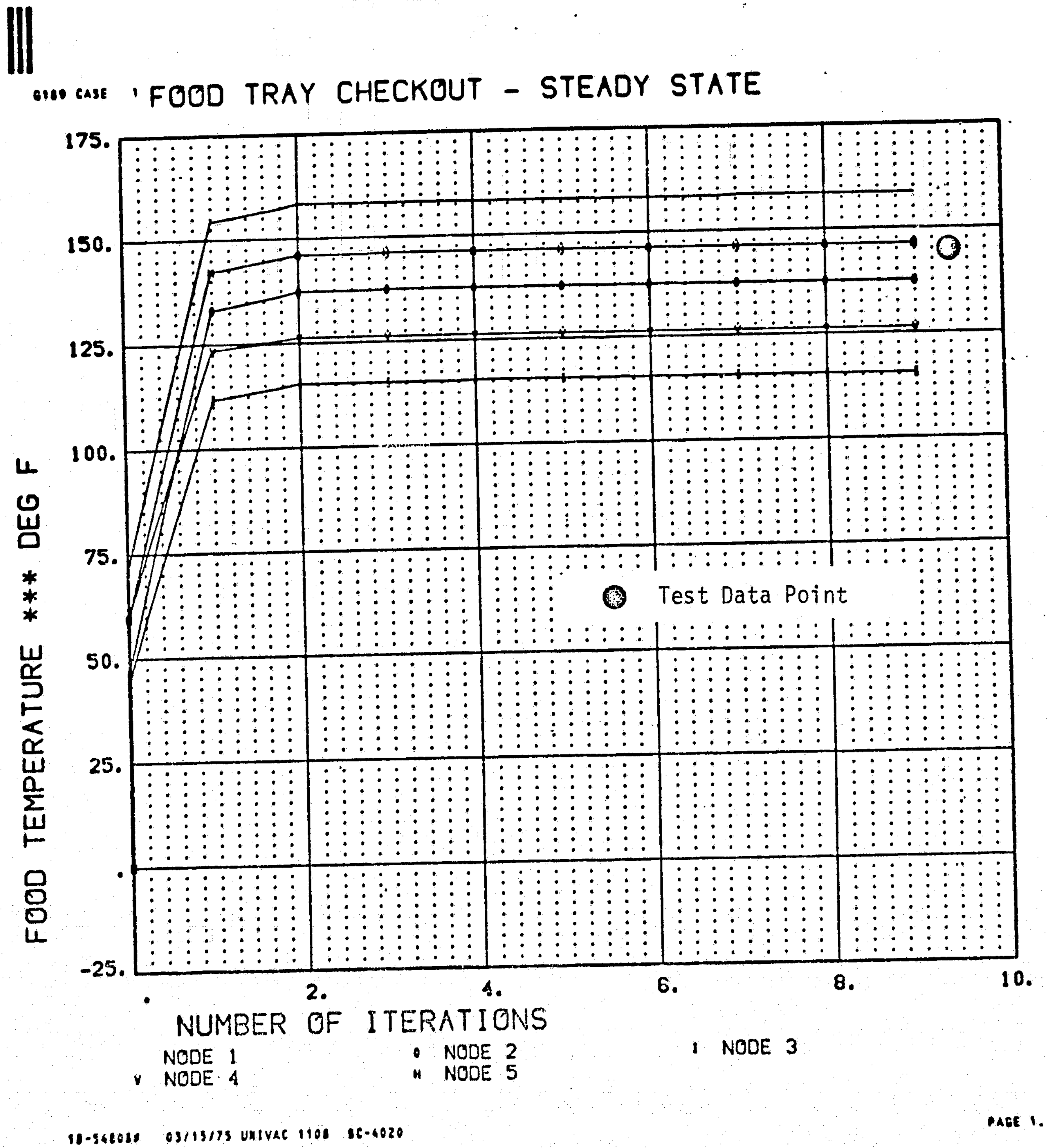

Figure 4-17. Temperatures for Food Tray Steady-State Checkout Run

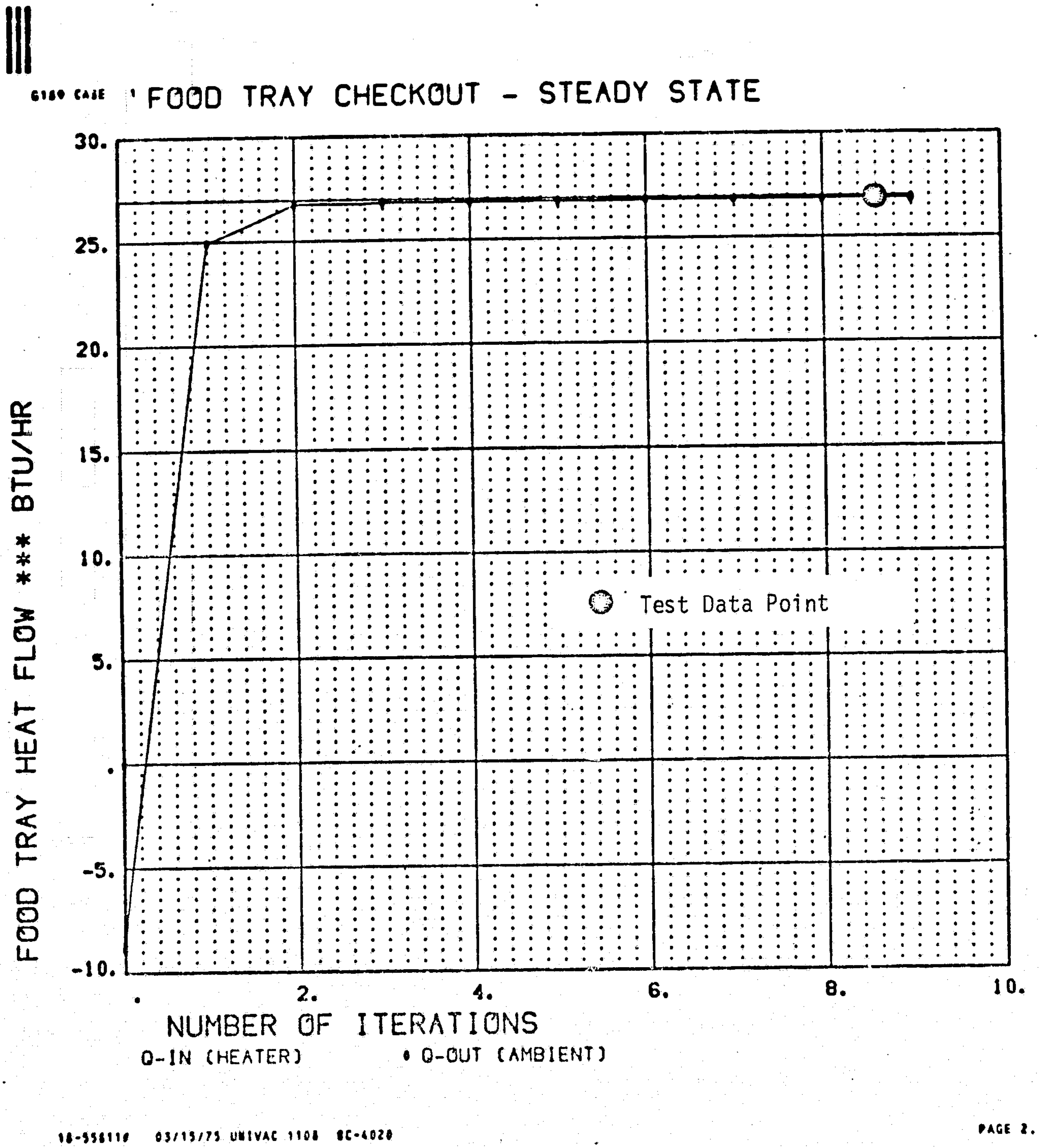

Figure 4-18. Heat Flow for Food Tray Steady-State Checkout Run

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TABLE 4-9

FINAL TRANSIENT SOLUTION FOR FOOD TRAY CHECKOUT RUN

COMP NO 2 FTRAY SUBR NO 66 PRI SOR 0 SEC SOR 0 COMP PASS NO 112 TIME = 10080.0 SEC

FAILURE FLAGS -- COMP= 0 LOOP= 0

A(1)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	

CPA= .00000 WTMA= .00000 RHOA= .00000 VISCA= .00000 XKA= .00000

B(1)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	

CPB= .00000 WTMB= .00000 RHOB= .00000 VISCB= .00000 XKB= .00000

R(1)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(11)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(21)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(31)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(41)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(51)=	137.37	.43379	26.955	70.000	.14800	8.7295	70.000	.62500-01	4.5657	70.000
R(61)=	.21600	12.767	135.08	.00000	35.485	1.0000	1.8750	1.1250	7.9000	.62489
R(71)=	160.88	136.87	115.12	127.15	146.81	.11379	.25566	.22361	.25544	.21818
R(81)=	.56364	.99000	.25000	157.00	2.0000	.78000	.20290	1.9000	.72400	52.000
R(91)=	1.1371	.66182-01	.66182-01	.48073	.48073	.00000	.00000	.00000	.00000	.00000
R(101)=	59.000	.23418-01	.61430-01	1.0000	1.0000	.00000	.00000	.00000	.00000	.00000

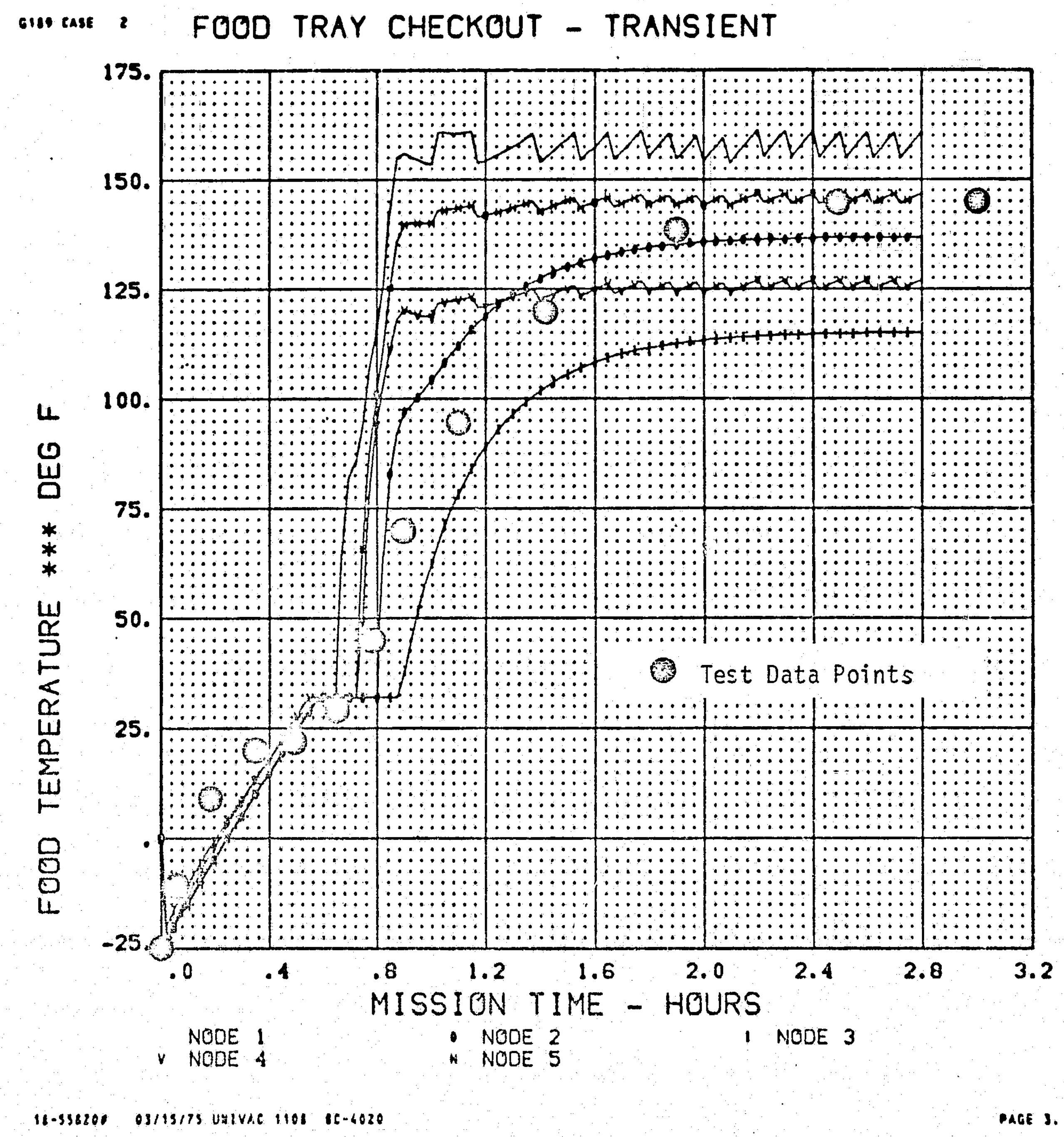

Figure 4-19. Temperatures for Food Tray Transient Checkout Run

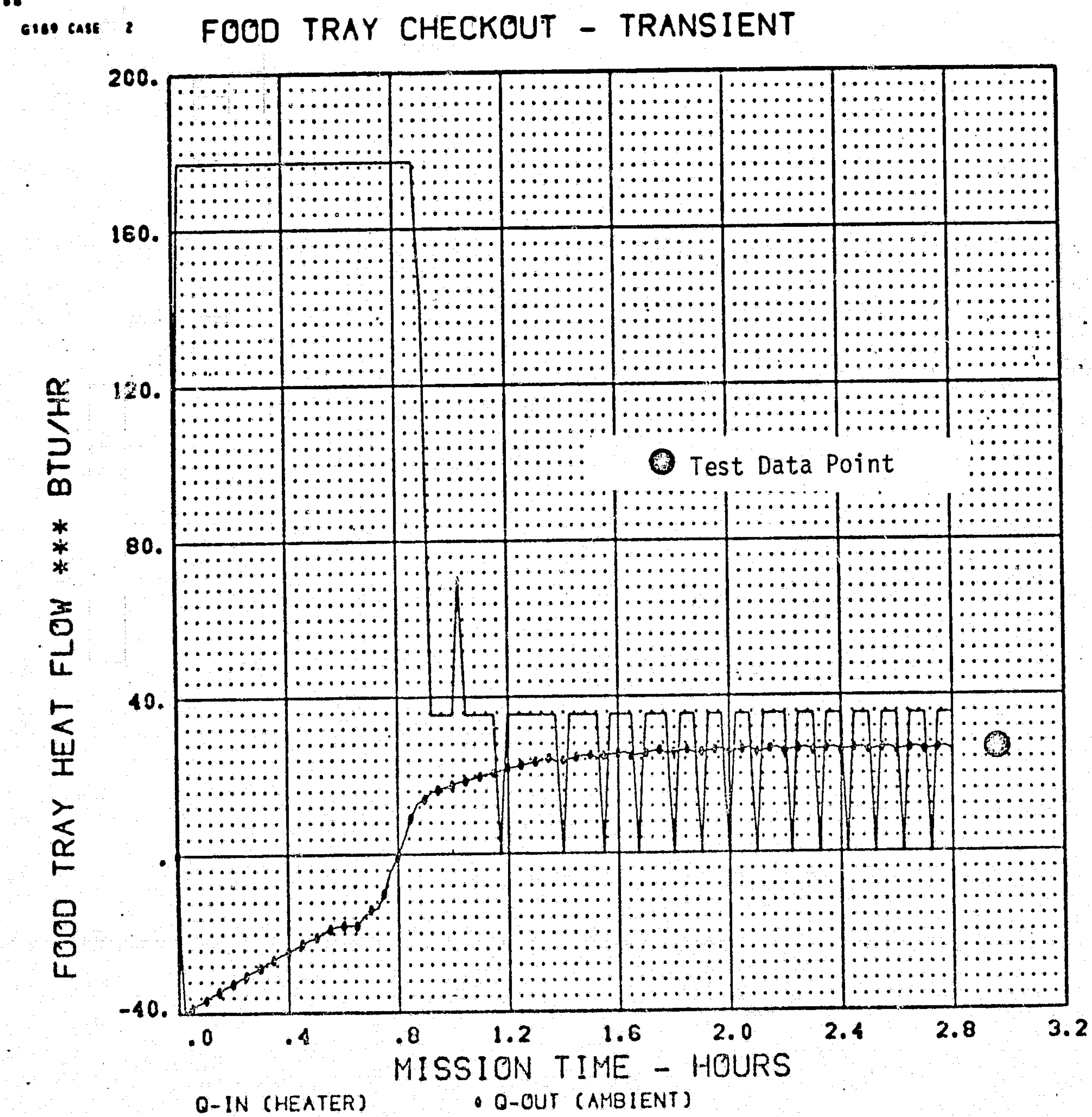

Figure 4-20. Heat Flow for Food Tray Transient Checkout Run

4.2.2 (Continued)

range. The heat flow into and out of the food is shown in Figure 4-20. After 3 hours warming time, the total heat dissipated to ambient was 7.9 watts (26.95 Btu/hr), which was also equal to the average heater input power. The average equilibrium heater power required in the tests of Reference 23 was also 79 watts, thus indicating good correlation between the G-189A model and Skylab food tray performance.

4.3 ROSMOS CHECKOUT RUN

The final design of the reverse osmosis unit to be selected for future space missions has not yet been determined, and consequently experimental data could not be used for comparison. However, a conceptual design was assumed for analytical simulation with currently feasible model data used as input. Two cases were run: one steady state and one transient. The component input data for the two cases were identical and are listed in Table 4-10. These data correspond to the subroutine input default data listed in Table 3-9 except that a thermal conduction path between the unit and ambient surroundings has been added in R-array locations 54 through 64. In its planned mode of operation, the reverse osmosis unit will operate near ambient temperature, so no thermal path was included in the default data. For this case, the fluid outlet temperature is equal to its inlet temperature. However, some unit designs require elevated temperatures, so this case was run to check out the subroutine thermal balance.

4.3.1 ROSMOS Steady-State Case

The final component data for the steady-state reverse osmosis unit simulation are listed in Table 4-11. The results for the run are plotted in Figures 4-21 through 4-23. The outlet brine flow rate should be computed as described in Paragraph 3.3, equations (3.3.7), (3.3.9), and (3.3.10). For the input data assumed for the component, these calculations should yield:

$$\Delta P_{osmosis} = R(75)\left[\frac{R(72)}{R(71)}\right] = 200\left[\frac{0.008}{0.01}\right] = 160 \text{ psia}$$

$$\dot{m}_{product_{design}} = R(70)\ R(65) = (9)(0.96) = 8.64$$

TABLE 4-10

G-189A INPUT DATA FOR REVERSE OSMOSIS UNIT CHECKOUT RUN

KBAS	7	0 69 20		0	1	0	1	18
KARY	7	16	2					
VARY	7	51 75.		R/O UNIT TEMP (F)				
VARY	7	54 75.		AMBIENT GAS TEMP				
VARY	7	55 3.		UA, AMBIENT GAS-TO-INSULATION SURFACE				
VARY	7	57 75.		AMBIENT WALL TEMP (F)				
VARY	7	58 .3		RADIATION CONDUCTOR, WALL-TO-OUTER SURFACE				
VARY	7	60 75.		ATTACHED STRUCTURAL TEMP (F)				
VARY	7	61 .4		CONDUCTOR, STRUCTURE-TO-R/O UNIT				
VARY	7	63 75.		INSULATION OUTER SURFACE TEMP (F)				
VARY	7	64 .12		THERMAL CONDUCTOR THROUGH INSULATION				
VARY	7	65 .96		DESIGN RECOVERY FACTOR (WP/WF)				
VARY	7	66 .93		OVERALL SALT REJECTION FACTOR-DESIGN				
VARY	7	67 50.		DESIGN PRODUCT WATER PRESSURE (PSIA)				
VARY	7	70 9.		DESIGN FEEDWATER FLOW, LB/HR				
VARY	7	71 .01		DESIGN FEEDWATER TOTAL SALT CONCENTRATION				
VARY	7	72 .008		ACTUAL FEEDWATER TOTAL SALT CONCENTRATION				
VARY	7	73 900.		DESIGN FEEDWATER PRESSURE (PSIA)				
VARY	7	74 45.		ACTUAL PRODUCT WATER PRESSURE (PSIA)				
VARY	7	75 200.		OVERALL DESIGN AVERAGE OSMOTIC PRESSURE (PSIA)				
VARY	7	76 2.5		THERMAL CAPACITANCE OF UNIT - DRY				
VARY	7	77 .0004						
VARY	7	78 .00004						
VARY	7	79 .0005						
VARY	7	80 .00003						
VARY	7	81 .97						
VARY	7	82 .70						

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TABLE 4-11

FINAL STEADY-STATE SOLUTION FOR REVERSE OSMOSIS UNIT CHECKOUT RUN

```
 COMP NO     7  ROSMOS     SUBR NO  69     PRI SOR      0     SEC SOR      0          COMP PASS NO     14       TIME = 21600.0 SEC
                           FAILURE FLAGS -- COMP=   0      LOOP=  0

   A(1)=   10.200     165.00     .00000     940.00     .00000     .00000     .00000     .00000     .00000     .00000
           .00000     .00000     .00000     .00000     .00000     .00000     .00000     .00000     .00000
                CPA=  1.0000           WTMA=  18.000           RHOA=  62.400          VISCA=  1.4000          XKA=  .36000
   B(1)=   .00000     .00000     .00000     .00000     .00000     .00000     .00000     .00000     .00000     .00000
           .00000     .00000     .00000     .00000     .00000     .00000     .00000     .00000     .00000
                CPB=  .00000           WTMB=  .00000           RHOB=  .00000          VISCB=  .00000          XKB=  .00000
```

```
 COMP NO     7  ROSMOS     SUBR NO  69     PRI SOR      0     SEC SOR      0          COMP PASS NO     14       TIME = 21600.0 SEC
                           FAILURE FLAGS -- COMP=   0      LOOP=  0          ELAPSED TIME IN MILLISEC -- CP=  113      PP=     0
 VR(  1)=  .43015      VR(  2)=  160.67      VR(  3)=  940.00      VR(  4)=  940.00      VR( 20)=  9.7698      VR( 21)=  160.67
 VR( 22)=  45.000      VR( 23)=  45.000      VR( 51)=  160.67      VR( 52)=  .51581      VR( 53)=  44.188      VR( 54)=  75.000
 VR( 55)=  3.0000      VR( 56)=  8.9735      VR( 57)=  75.000      VR( 58)=  .30000      VR( 59)=  .94790      VR( 60)=  75.000
 VR( 61)=  .40000      VR( 62)=  34.267      VR( 63)=  77.991      VR( 64)=  .12000      VR( 65)=  .96000      VR( 66)=  .93000
 VR( 67)=  50.000      VR( 68)=  .17645      VR( 69)=  .58333-03   VR( 70)=  9.0000      VR( 71)=  .10000-01   VR( 72)=  .80000-02
 VR( 73)=  900.00      VR( 74)=  45.000      VR( 75)=  200.00      VR( 76)=  2.5000      VR( 77)=  .40000-03   VR( 78)=  .40000-04
 VR( 79)=  .50000-03   VR( 80)=  .30000-04   VR( 81)=  .97000      VR( 82)=  .70000      VR( 83)=  .15625-04   VR( 84)=  .93750-05
 VR( 85)=  .11501-01   VR( 86)=  .49844-03   VR( 87)=  165.00      VR( 88)=  10.200      VR( 89)=  44.189      VR( 90)=  .64373-03
 VR( 91)=  .00000      VR( 92)=  .00000      VR( 93)=  .00000      VR( 94)=  .00000      VR( 95)=  .00000      VR( 96)=  .00000
                       CPP=  1.0000          WTMP=  18.000         RHOP=  62.400         VISCP=  1.4000        XKP=  .36000
                       CPS=  1.0000          WTMS=  18.000         RHOS=  62.400         VISCS=  1.4000        XKS=  .36000
```

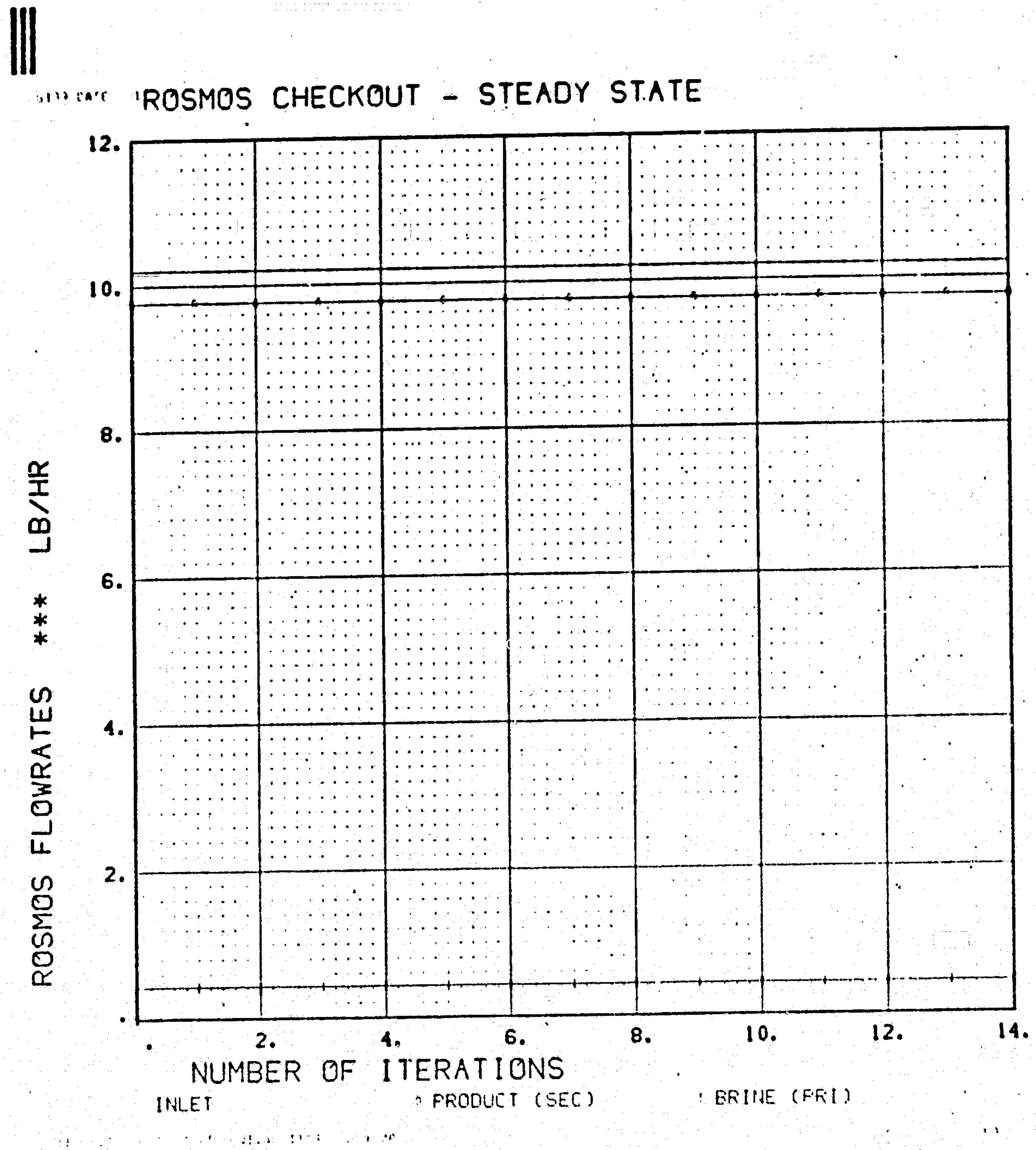

Figure 4-21. Flow Rates for Reverse Osmosis Unit Steady-State Checkout Run

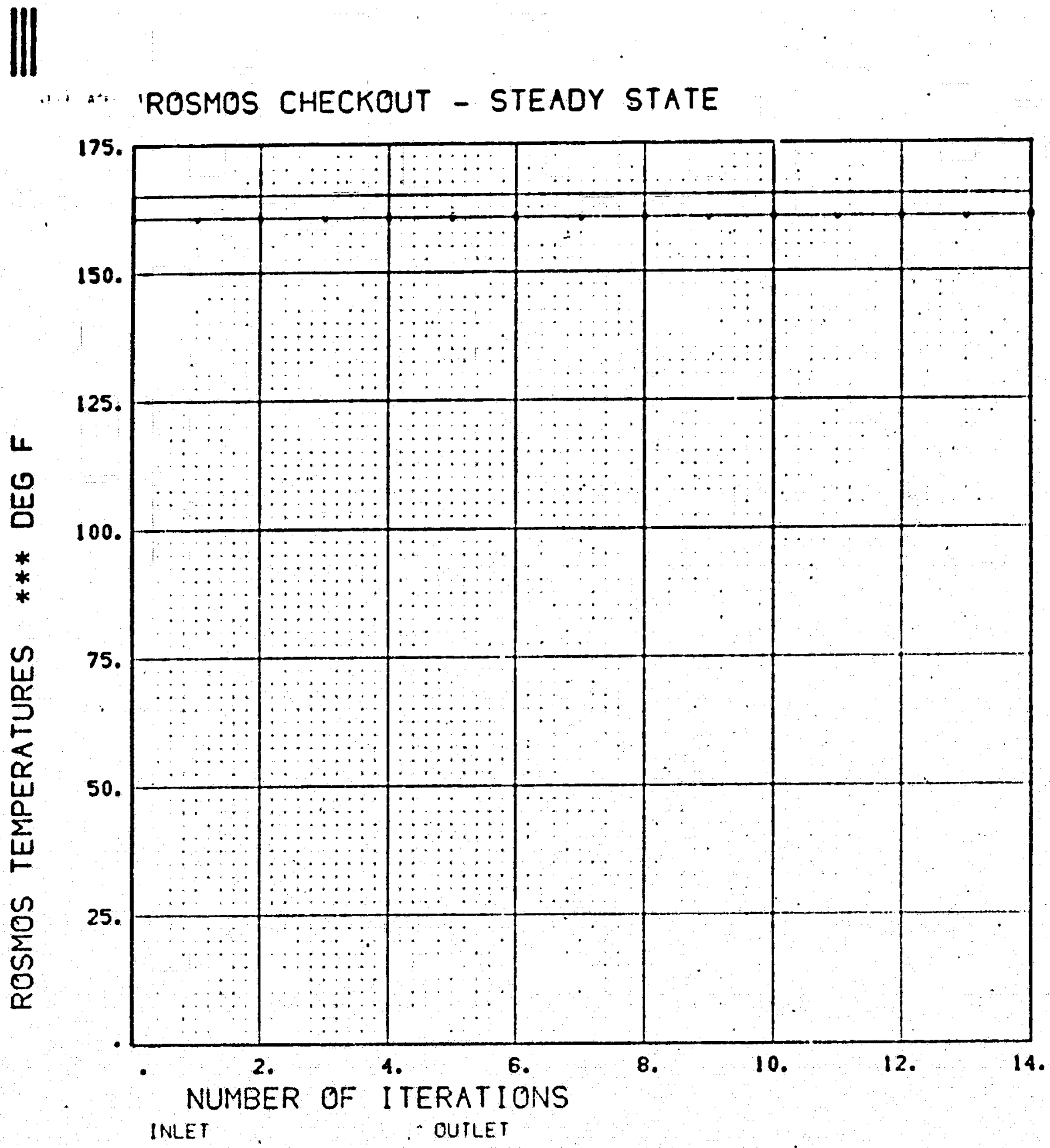

Figure 4-22. Temperatures for Reverse Osmosis Unit Steady-State Checkout Run

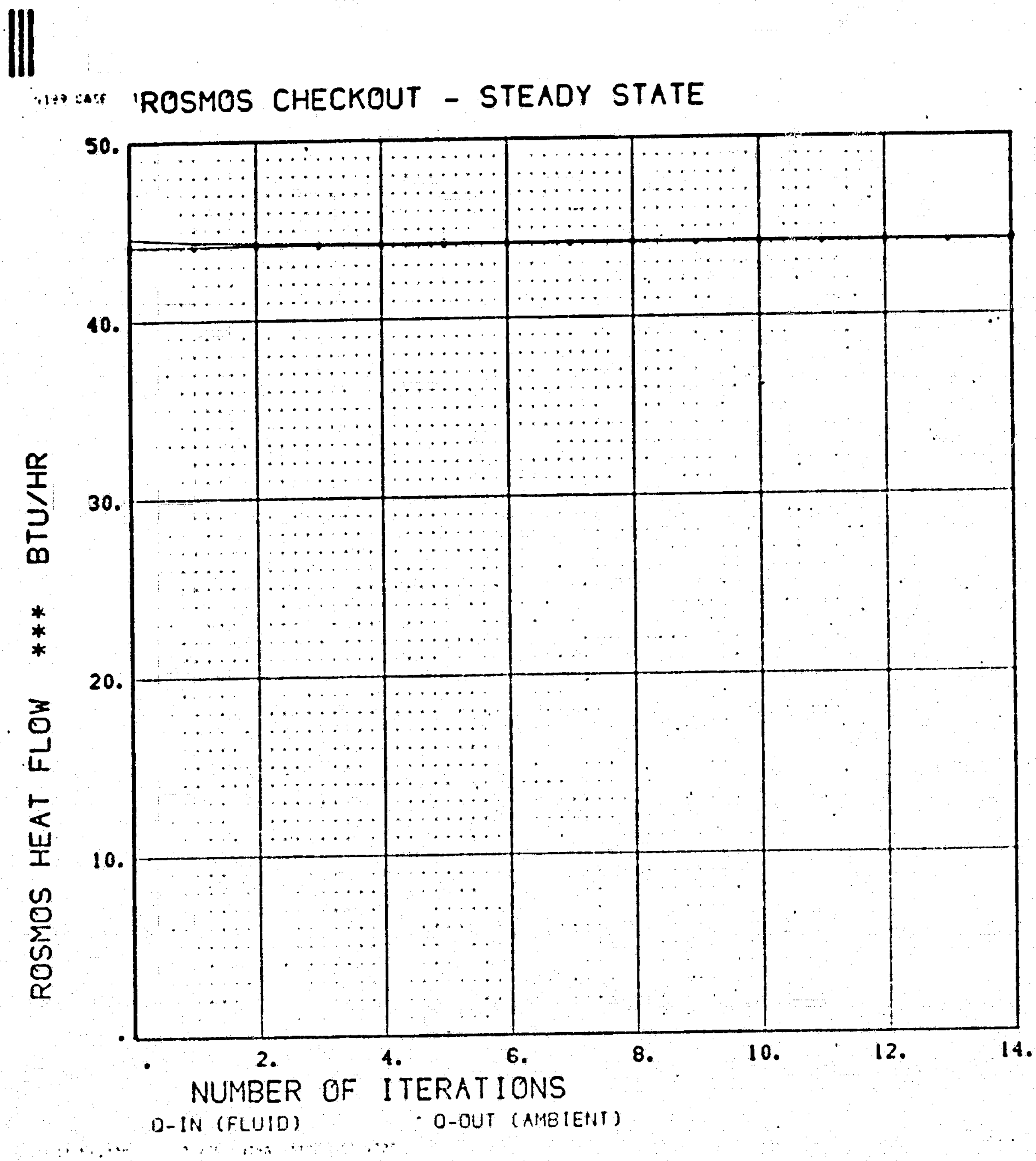

Figure 4-23. Heat Flow for Reverse Osmosis Unit Steady-State Checkout Run

4.3.1 (Continued)

$$\dot{m}_{product_{actual}} = \dot{m}_{product_{design}} \left[\frac{A(4) - R(74) - \Delta P_{osmosis}}{R(73) - R(67) - R(75)}\right]$$

$$= 8.64 \left[\frac{940 - 45 - 160}{900 - 50 - 200}\right] = 9.770 \text{ lb/hr}$$

The product flow rate thus calculated agrees exactly with the flow plotted in Figure 4-21 and the actual value [R(20)] shown in Table 4-11. The brine flow is seen in Figure 4-21 and Table 4-11 to be the difference between the feed (inlet) and product flows as required.

The thermal data in Figure 4-23 show that, in steady-state conditions, the heat into the reverse osmosis unit (from the warm fluid flowing through it) is equal to the heat dissipated through the unit structure to ambient surroundings. The energy input from the fluid is

$$q_{fluid\ in} = \dot{m}\ c_p\ \Delta T = A(1) * CPA\ [A(2) - R(2)]$$

$$= (10.2 \text{ lb/hr})(1 \text{ Btu/lb °F})(165°F - 160.67°F) = 44.2 \text{ Btu/hr}$$

The energy out to ambient surroundings by conduction, convection, and radiation is given by

$$q_{out} = G_{effective}\ \Delta T = R(52)\ [R(51) - R(54)]$$

$$= (0.516 \text{ Btu/hr °F})(160.67°F - 75°F) = 44.2 \text{ Btu/hr}$$

Thus, the predicted reverse osmosis unit temperature agrees with the steady-state heat balance as required.

4.3.2 ROSMOS Transient Case

The final component data for the transient case are listed in Table 4-12. The results for 60 minutes of operation are plotted in Figures 4-24 through 4-26. The conditions assumed for the transient case were identical to those used in the steady-state case discussed in Paragraph 4.3.1. The 60 minutes operating time was sufficiently long to achieve near equilibrium conditions, and the final results may be compared directly with the steady-state solution in Table 4.11 and Figures 4-21 through 4-23. It is apparent in the figures and tabular data that the final transient and steady-state solutions are identical, as they should be.

4.4 SHOWER CHECKOUT RUN

The SHOWER subroutine was used to model the zero gravity whole body shower described in References 19 and 24. Only the conditions in the new shower stall component are simulated in the checkout run since the peripheral equipment (pump, fan, heater, etc.) can be modeled using standard G-189A subroutines already operational. Note that the complete shower subsystem is included in the Space Station model described in Section 6. A steady-state and transient case were run, as described in the following paragraphs. For both cases the subroutine input default data listed in Table 3-12 were used.

4.4.1 Steady-State Shower Case

The final solution for the steady-state shower run is given in Table 4-13. The results of the run are plotted in Figures 4-27 through 4-30. The inlet air and water flow rates and temperatures were set to actual test conditions. For the first 10 iterations, the shower water was turned on. Thus, the relative humidity in the stall was assumed in the model to be 100 percent as reflected in Figure 4-29. For the second 10 iterations, the inlet water flow was turned off, and evaporative drying within the stall was computed. In Figure 4-28 is shown the total heat balance for the shower stall. At steady-state, both for water-on and water-off conditions, it is apparent that the heat input is equal to the total heat

TABLE 4-12

FINAL TRANSIENT SOLUTION FOR REVERSE OSMOSIS UNIT CHECKOUT RUN

COMP NO, 7 ROSMOS SUBR NO 69 PRI SOR 0 SEC SOR 0 COMP PASS NO 43 TIME = 3990.0 SEC

FAILURE FLAGS -- COMP= 0 LOOP= 0

A(1)=	10.200	165.00	.00000	940.00	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPA=	1.0000	WTMA=	18.000	RHOA=	62.400	VISCA=	1.4000	XKA=	.36000
B(1)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPB=	.00000	WTMB=	.00000	RHOB=	.00000	VISCB=	.00000	XKB=	.00000

R(1)=	.43015	159.51	940.00	940.00	.00000	.00000	.00000	.00000	.00000	.00000
R(11)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	9.7698
R(21)=	159.51	45.000	45.000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(31)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(41)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(51)=	159.51	.51581	43.529	75.000	3.0000	8.8396	75.000	.30000	.93363	75.000
R(61)=	.40000	33.755	77.947	.12000	.96000	.93000	50.000	.17645	.58333-03	9.0000
R(71)=	.10000-01	.80000-02	900.00	45.000	200.00	2.5000	.40000-03	.40000-04	.50000-03	.30000-04
R(81)=	.97000	.70000	.15625-04	.93750-05	.11501-01	.49844-03	165.00	10.200	55.995	13.709
R(91)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000

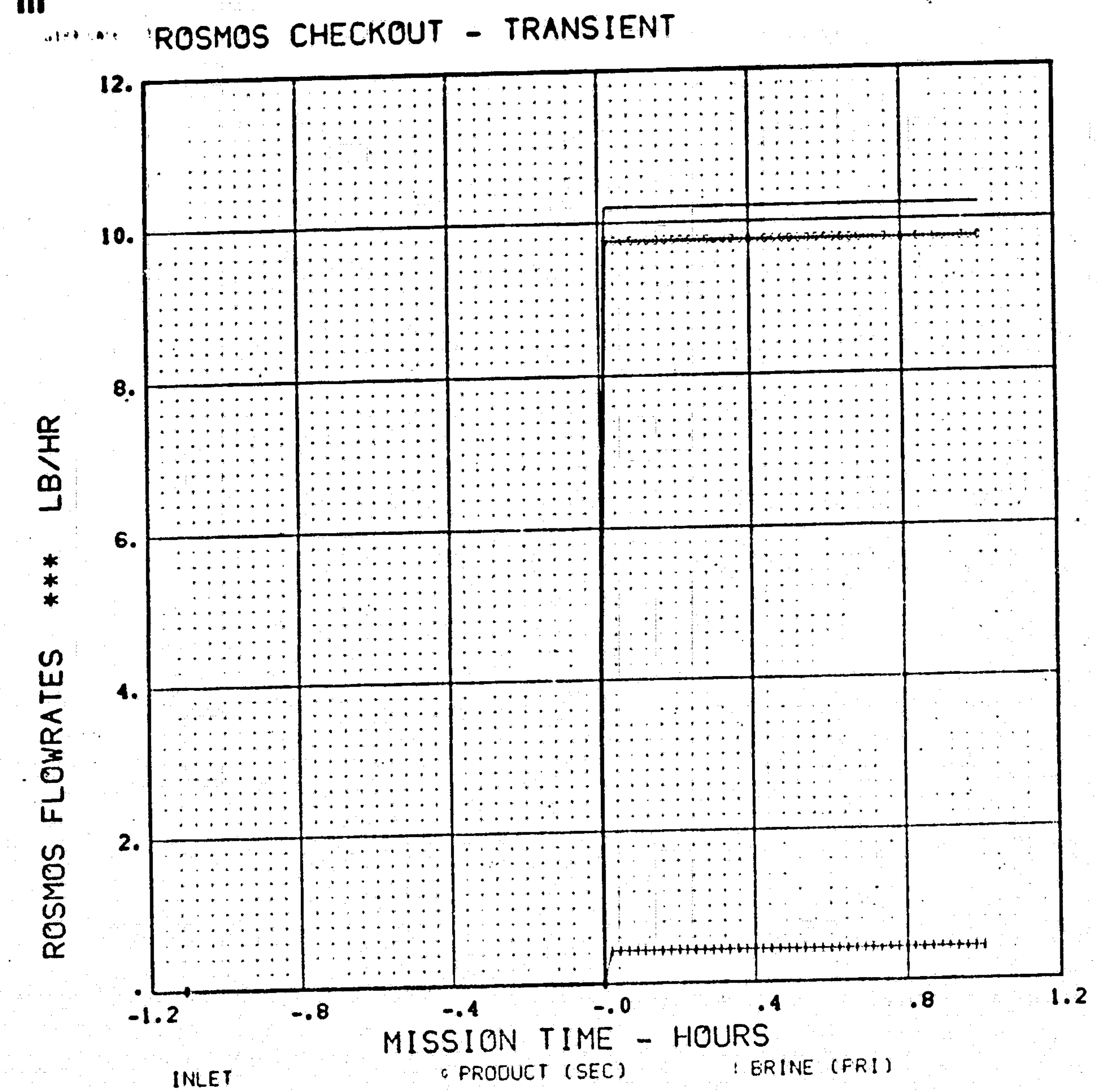

Figure 4-24. Flow Rates for Reverse Osmosis Unit Transient Checkout Run

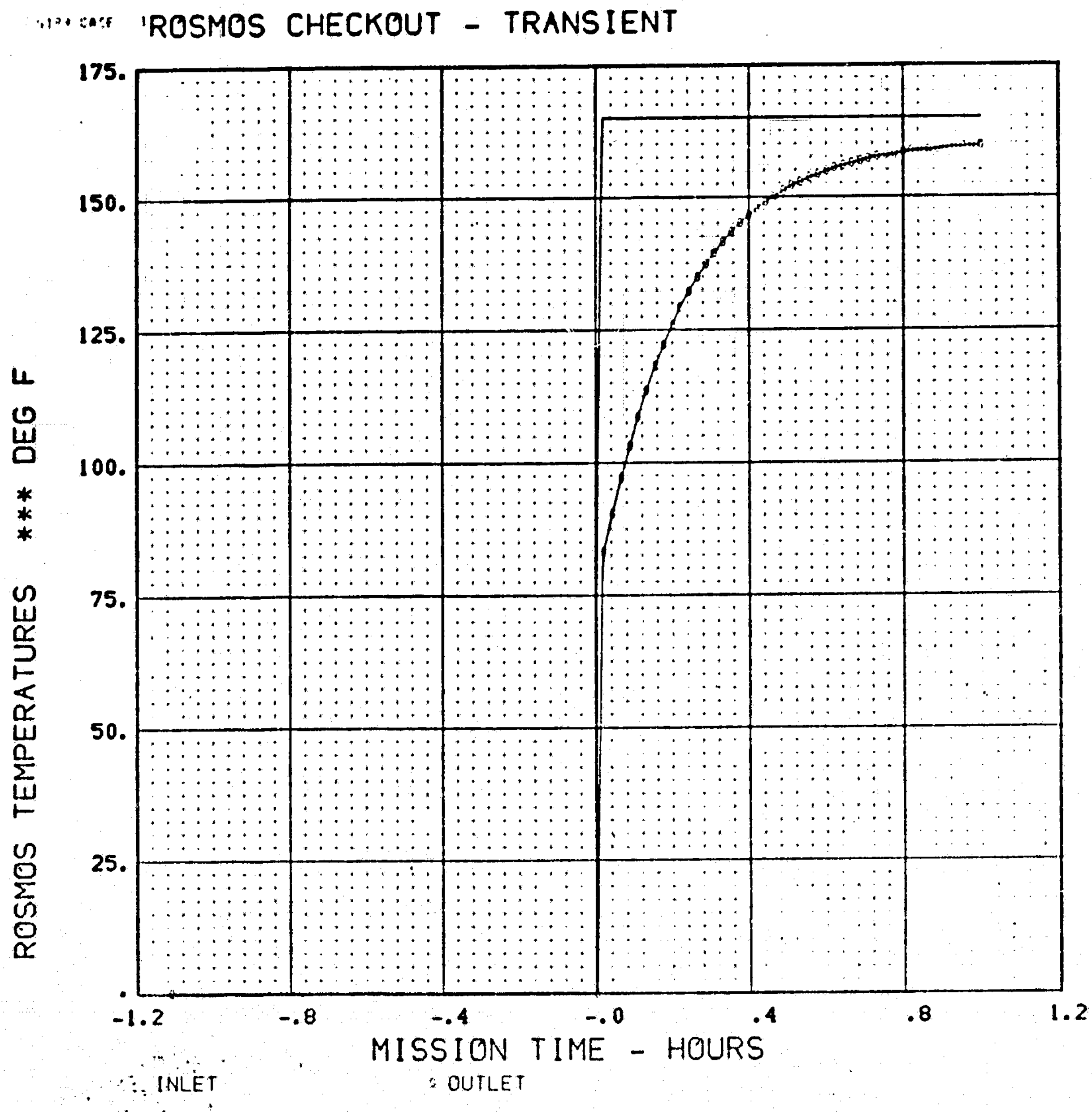

Figure 4-25. Temperatures for Reverse Osmosis Unit Transient Checkout Run

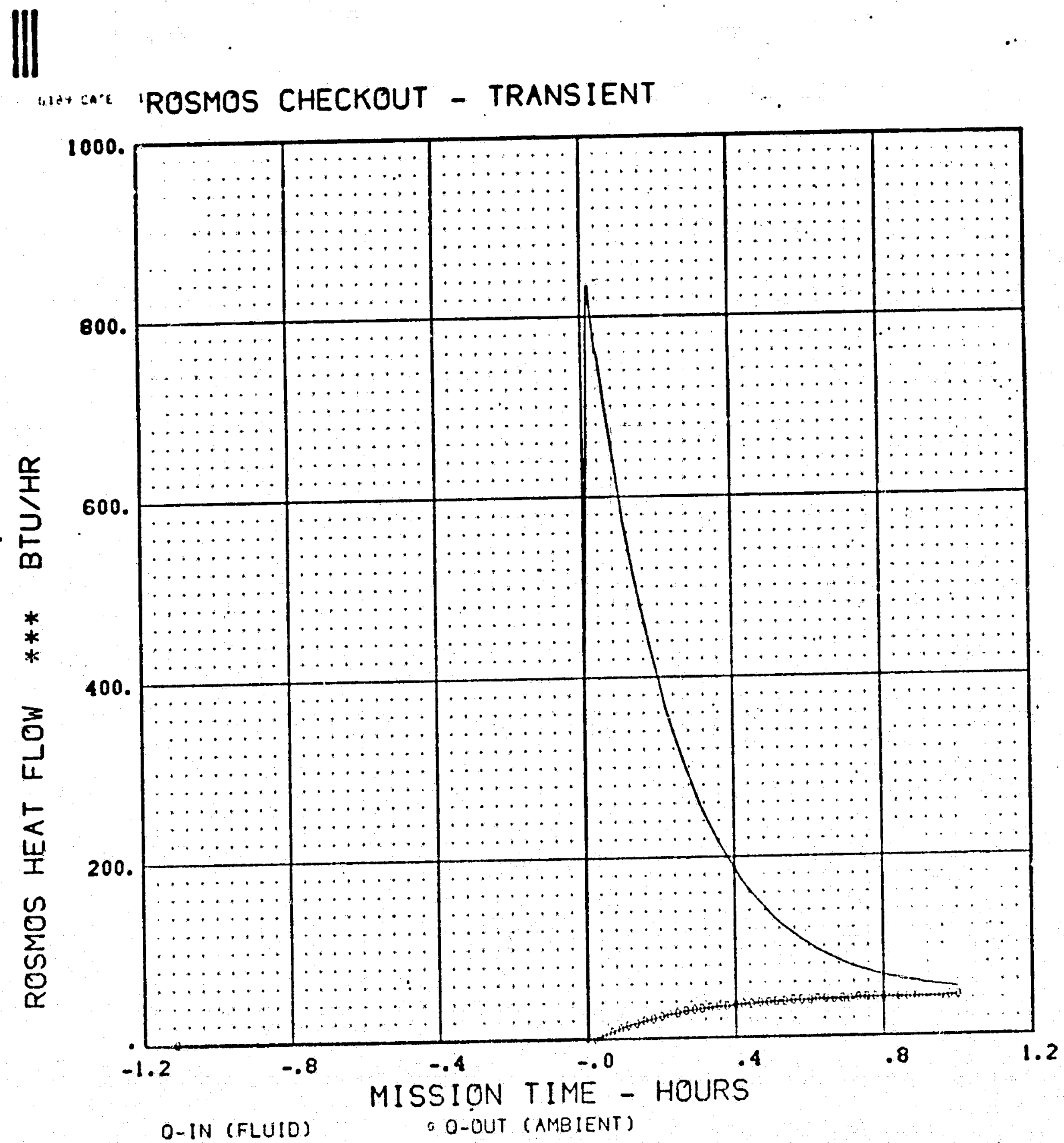

Figure 4-26. Heat Flow for Reverse Osmosis Unit Transient Checkout Run

TABLE 4-13(a)

FINAL STEADY-STATE SOLUTION FOR SHOWER CHECKOUT RUN WITH WATER TURNED ON

A(1)=	126.00	105.00	14.700	14.700	123.48	2.5200	.00000	.24124	28.888	28.980
	94.020	.48000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPA=	.24522	WTMA=	28.543	RHOA=	.69215-01	VISCA=	.47000-01	XKA=	.15400-01
B(1)=	160.00	105.00	45.000	45.000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPB=	1.0000	WTMB=	18.000	RHOB=	62.400	VISCB=	1.4000	XKB=	.36000

VR(1)=	306.03	VR(2)=	93.733	VR(3)=	14.700	VR(4)=	14.700	VR(5)=	123.51	VR(6)=	4.3530
VR(7)=	178.17	VR(8)=	.24122	VR(9)=	28.896	VR(10)=	28.874	VR(11)=	94.020	VR(12)=	.61087
VR(13)=	.00000	VR(14)=	.00000	VR(20)=	.00000	VR(21)=	.00000	VR(22)=	.00000	VR(23)=	.00000
VR(51)=	93.249	VR(52)=	59.096	VR(53)=	1196.8	VR(54)=	70.000	VR(55)=	14.000	VR(56)=	283.51
VR(57)=	70.000	VR(58)=	14.000	VR(59)=	305.71	VR(60)=	70.000	VR(61)=	30.000	VR(62)=	607.53
VR(63)=	90.251	VR(64)=	.00000	VR(65)=	.00000	VR(66)=	650.00	VR(67)=	.13087	VR(68)=	.90000
VR(69)=	39.000	VR(70)=	.10575	VR(71)=	13.000	VR(72)=	2.3902	VR(73)=	3.0053	VR(74)=	4.5000
VR(75)=	1.0000	VR(76)=	24.000	VR(77)=	97.217	VR(78)=	16.575	VR(79)=	.00000	VR(80)=	1.0000
VR(81)=	46.000	VR(82)=	1.0000	VR(83)=	.00000	VR(84)=	.00000	VR(85)=	.00000	VR(86)=	.00000
VR(87)=	.30000	VR(88)=	.78720	VR(89)=	3.0522	VR(90)=	.99912	VR(91)=	731.89	VR(92)=	-466.91
VR(93)=	1196.8	VR(94)=	1.9009	VR(95)=	105.00	VR(96)=	.00000	VR(97)=	.00000	VR(98)=	.00000
VR(99)=	.00000	VR(100)=	.00000	VR(							

TABLE 4-13(b)

FINAL STEADY-STATE SOLUTION FOR SHOWER CHECKOUT RUN WITH WATER TURNED OFF

A(1)=	126.00	105.00	14.700	14.700	123.48	2.5200	.00000	.24124	28.888	2[illegible].980
	94.020	.48000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPA=	.24522	WTMA=	28.543	RHOA=	.69215-01	VISCA=	.47000-01	XKA=	.15400-01
B(1)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPB=	.00000	WTMB=	.00000	RHOB=	.00000	VISCB=	.00000	XKB=	.00000

VR(1)=	126.55	VR(2)=	87.810	VR(3)=	14.700	VR(4)=	14.700	VR(5)=	123.51	VR(6)=	3.0462
VR(7)=	.00000	VR(8)=	.24122	VR(9)=	28.896	VR(10)=	28.874	VR(11)=	94.020	VR(12)=	.61087
VR(13)=	.00000	VR(14)=	.00000	VR(20)=	.00000	VR(21)=	.00000	VR(22)=	.00000	VR(23)=	.00000
VR(51)=	82.154	VR(52)=	58.756	VR(53)=	714.03	VR(54)=	70.000	VR(55)=	14.000	VR(56)=	170.13
VR(57)=	70.000	VR(58)=	14.000	VR(59)=	179.32	VR(60)=	70.000	VR(61)=	30.000	VR(62)=	364.57
VR(63)=	82.152	VR(64)=	.00000	VR(65)=	.00000	VR(66)=	650.00	VR(67)=	.13087	VR(68)=	.90000
VR(69)=	39.000	VR(70)=	.10575	VR(71)=	13.000	VR(72)=	2.3902	VR(73)=	3.0053	VR(74)=	4.5000
VR(75)=	1.0000	VR(76)=	24.000	VR(77)=	95.980	VR(78)=	16.575	VR(79)=	.80000	VR(80)=	1.0000
VR(81)=	46.000	VR(82)=	1.0000	VR(83)=	.52622	VR(84)=	.00000	VR(85)=	.00000	VR(86)=	.00000
VR(87)=	.00000	VR(88)=	.55974	VR(89)=	3.0522	VR(90)=	.85427	VR(91)=	731.89	VR(92)=	17.199
VR(93)=	714.03	VR(94)=	1.9009	VR(95)=	105.00	VR(96)=	.00000	VR(97)=	.00000	VR(98)=	.00000
VR(99)=	.00000	VR(100)=	.00000	VR(							

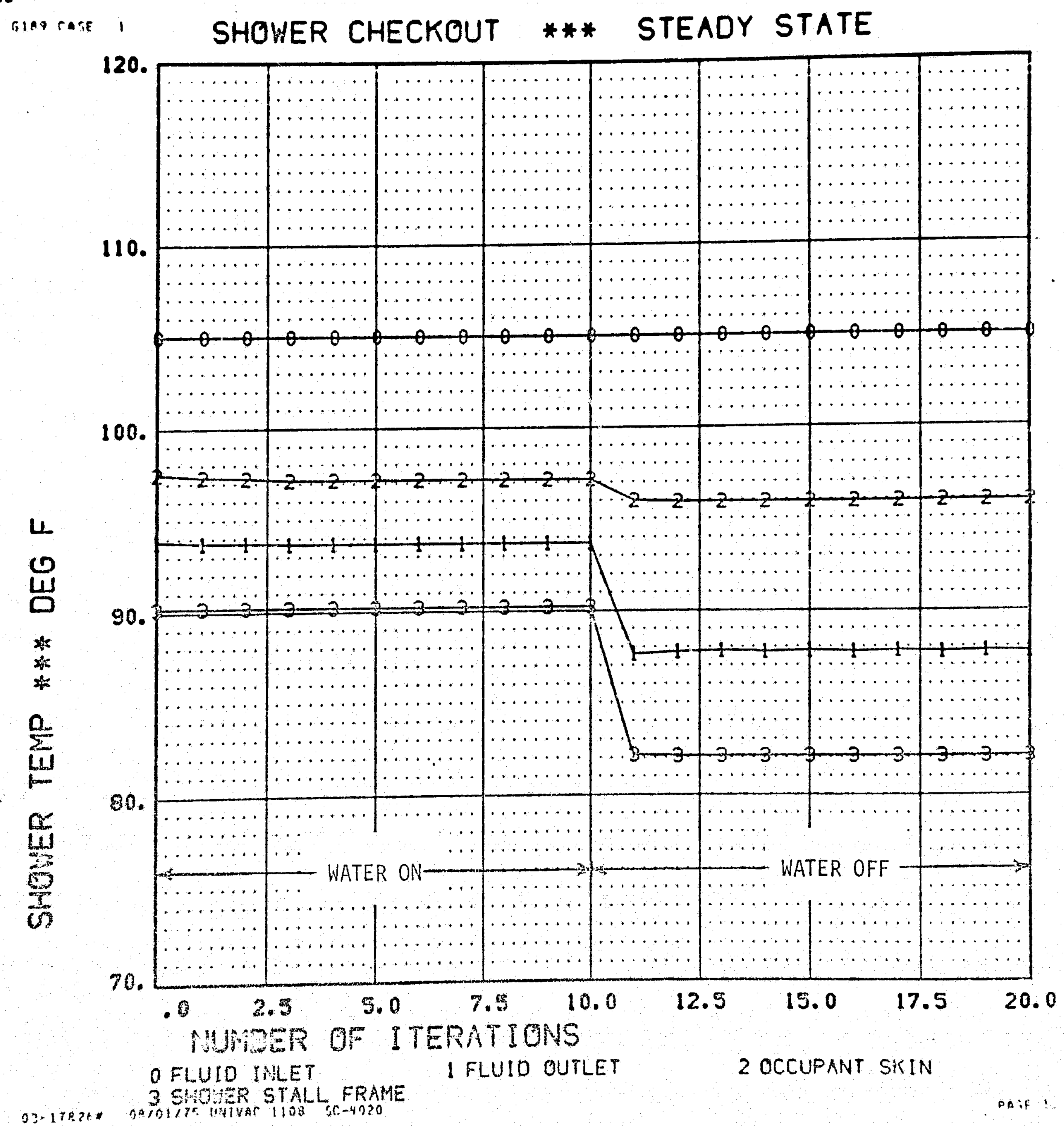

Figure 4-27. Temperatures for Shower Steady-State Checkout Run

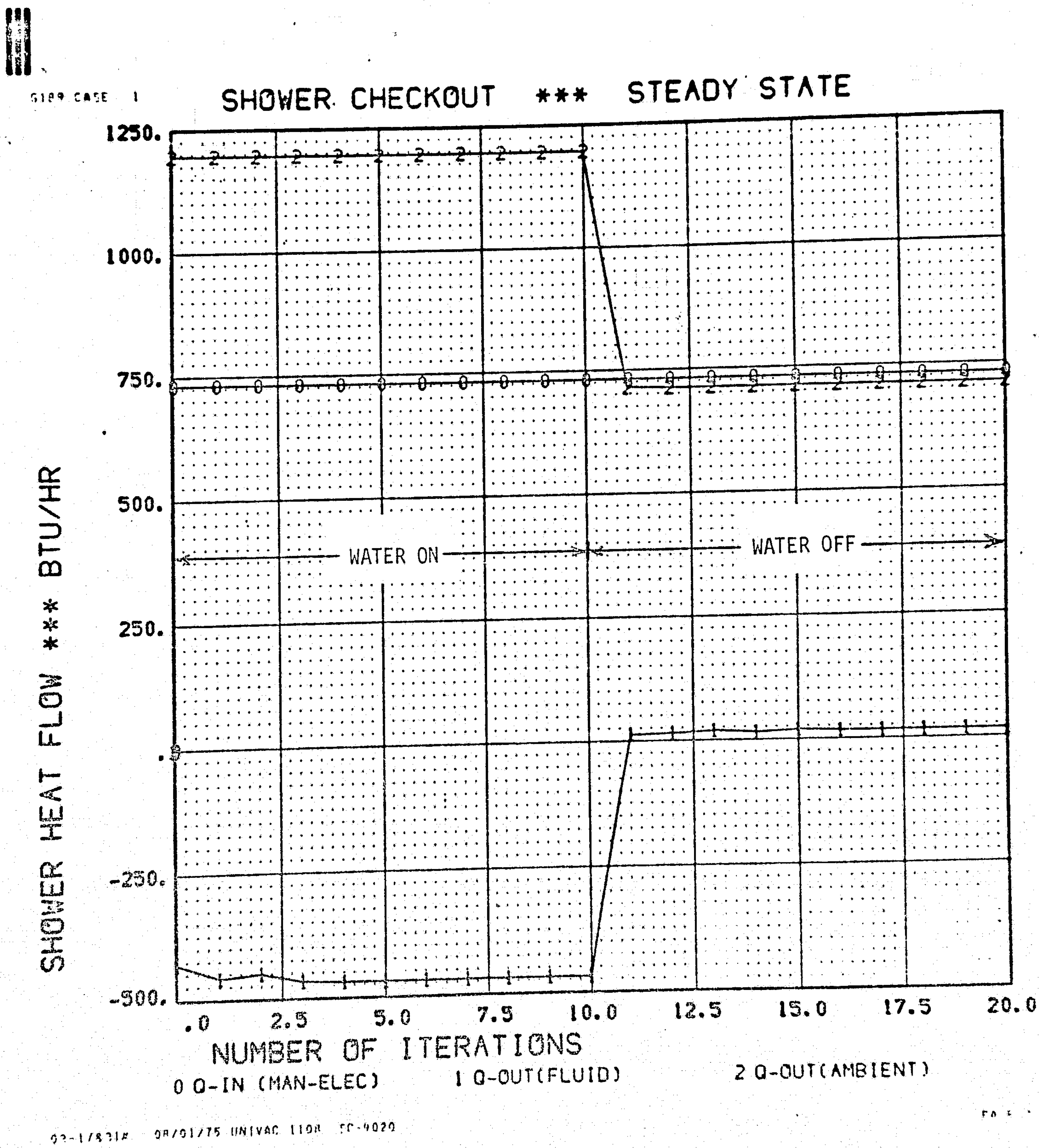

Figure 4-28. Heat Flow for Shower Steady-State Checkout Run

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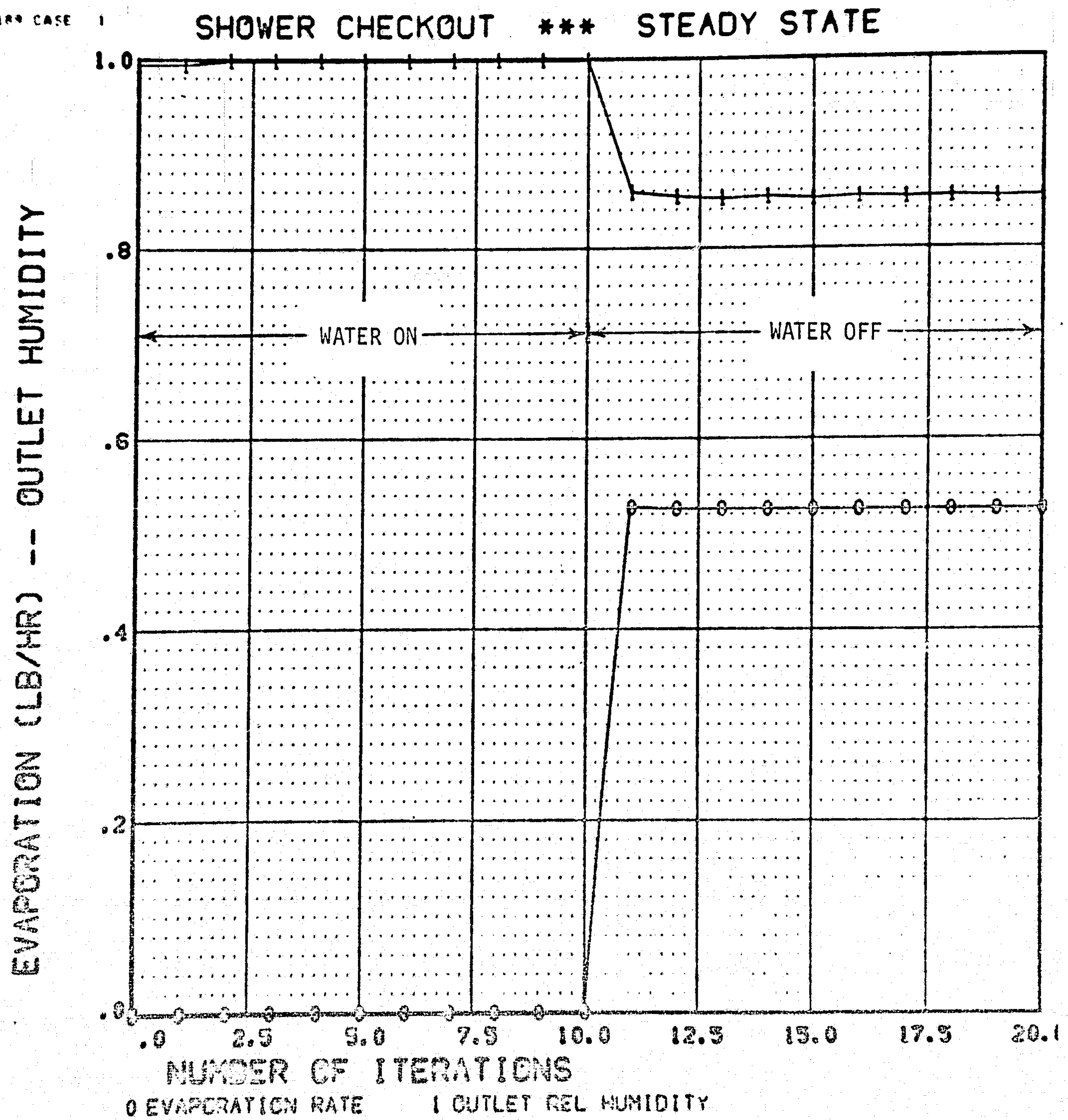

Figure 4-29. Evaporation Parameters for Shower Steady-State Checkout Run

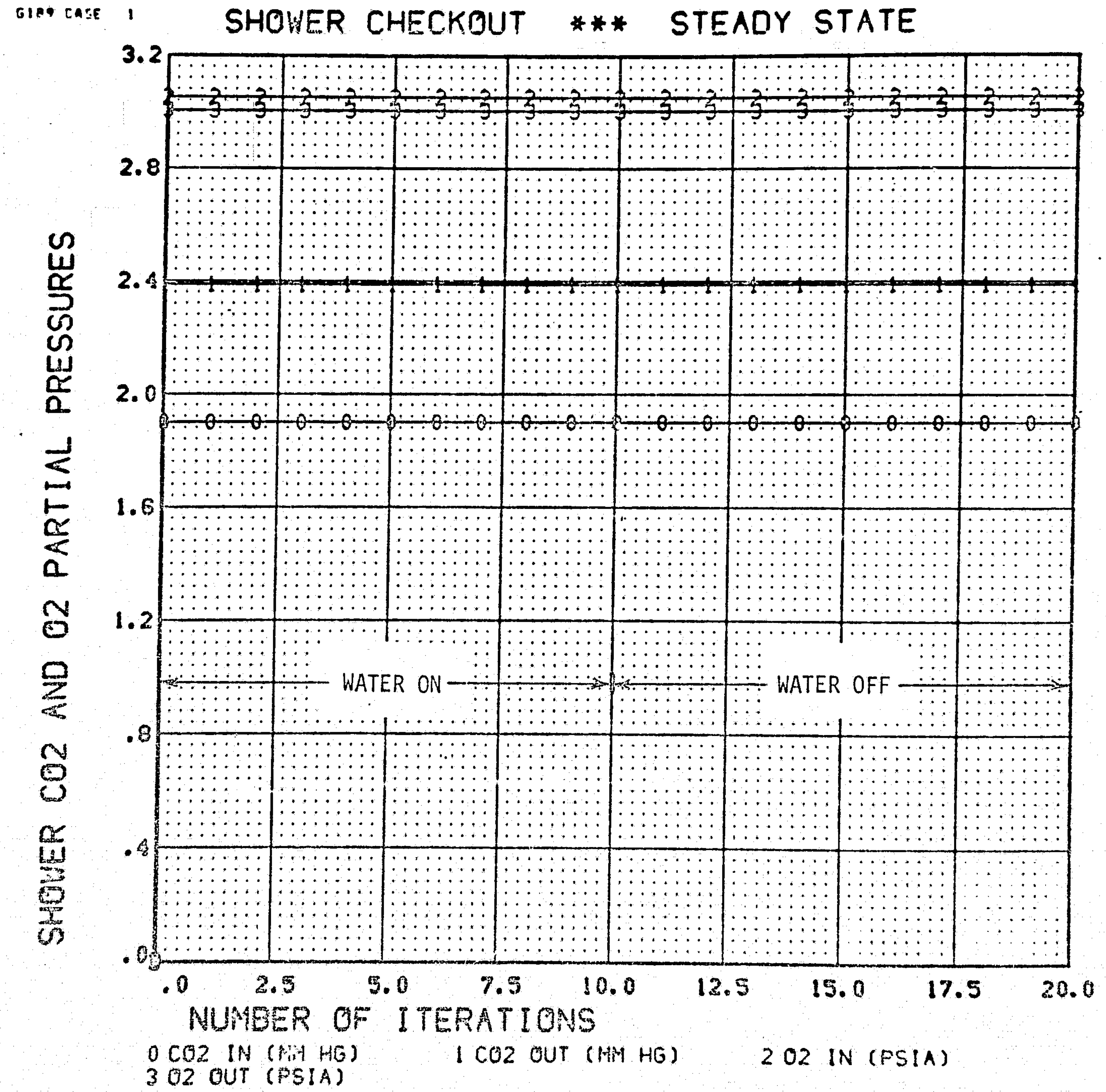

Figure 4-30. CO_2 and O_2 Levels for Shower Steady-State Checkout Run

4.4.1 (Continued)

out as required. The oxygen and CO_2 pressures within the stall, in Figure 4-30, agree with the computed values from equations (3.4.29) through (3.4.32). The shower temperatures, Figure 4-27, were checked by hand calculations and found to agree with the thermal balance given in Paragraph 3.4.3 for the walls and occupant nodes. During the air drying period, with water turned off, the results in Figure 4-29 show an evaporation rate slightly over 0.5 lb/hr. This value agreed favorably with limited test results in Reference 19. However, additional test data would be desirable to further refine and verify this result. The subroutine may be readily modified to fit other data using the correlation factor c_M discussed in Section 3.4.3.2.

4.4.2 Transient Shower Case

The G-189A input data for the transient shower run were identical to the steady-state case. For the first 1½ hours, the water was turned on and the remainder of the time it was off. Final solutions for the water-on and water-off conditions are given in Table 4-14. The results for the run are plotted in Figures 4-31 through 4-34. It is evident in the plots that both steady-state and transient runs reach the same final solution as required.

4.5 WASDRY CHECKOUT RUN

The WASDRY subroutine was used to model a clothes and dishes washer/dryer drum. The peripheral equipment (pump, fan, accumulator, etc.) were not included in the checkout runs described here since they can be modeled

TABLE 4-14(a)

FINAL TRANSIENT SOLUTION FOR SHOWER CHECKOUT RUN WITH WATER TURNED ON

A(I)=	126.00	105.00	14.700	14.700	123.48	2.5200	.00000	.24124	28.888	28.980
	94.020	.48000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPA=	.24522	WTMA=	28.543	RHOA=	.69215-01	VISCA=	.47000-01	XKA=	.15900-01
B(I)=	100.00	105.00	45.000	45.000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPB=	1.0000	WTMB=	18.000	RHOB=	62.400	VISCB=	1.4000	XKB=	.36000

R(1)=	306.03	93.706	14.700	14.700	123.51	4.3516	178.17	.24122	28.896	28.874
R(11)=	94.020	.61087	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(21)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(31)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(41)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
R(51)=	90.243	59.096	1196.3	70.000	14.000	283.40	70.000	14.000	305.59	70.000
R(61)=	30.000	607.30	90.243	.00000	.00000	650.00	.13087	.90000	39.000	.10575
R(71)=	13.000	2.3902	3.0053	4.5000	1.0000	24.000	97.169	16.575	.80000	1.0000
R(81)=	46.000	1.0000	.00000	10.000	170.00	10.000	.00000	.78698	3.0522	.99966
R(91)=	731.89	473.93	1196.3	1.9009	105.00	.00000	.00000	.00000	.00000	.00000

TABLE 4-14(b)

FINAL TRANSIENT SOLUTION FOR SHOWER CHECKOUT RUN WITH WATER TURNED OFF

A(1)=	126.00	105.00	14.700	14.700	123.48	2.5200	.00000	.24124	28.888	28.980
	94.020	.48000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPA=	.24522	WTMA=	28.543	RHOA=	.69215-01	VISCA=	.47000-01	XKA=	.15400-01
B(1)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPB=	.00000	WTMB=	.00000	RHOB=	.00000	VISCB=	.00000	XKB=	.00000

VR(1)=	126.56	VR(2)=	87.919	VR(3)=	14.700	VR(4)=	14.700	VR(5)=	123.51	VR(6)=	3.0532
VR(7)=	.00000	VR(8)=	.24122	VR(9)=	28.896	VR(10)=	28.874	VR(11)=	94.020	VR(12)=	.61087
VR(13)=	.00000	VR(14)=	.00000	VR(20)=	.00000	VR(21)=	.00000	VR(22)=	.00000	VR(23)=	.00000
VR(51)=	62.205	VR(52)=	58.758	VR(53)=	717.17	VR(54)=	70.000	VR(55)=	14.000	VR(56)=	170.88
VR(57)=	70.000	VR(58)=	14.000	VR(59)=	180.13	VR(60)=	70.000	VR(61)=	30.000	VR(62)=	366.16
VR(63)=	62.205	VR(64)=	.00000	VR(65)=	.00000	VR(66)=	650.00	VR(67)=	.13087	VR(68)=	.90000
VR(69)=	39.000	VR(70)=	.10575	VR(71)=	13.000	VR(72)=	2.3902	VR(73)=	3.0053	VR(74)=	4.5000
VR(75)=	1.0000	VR(76)=	24.000	VR(77)=	96.218	VR(78)=	16.575	VR(79)=	.60000	VR(80)=	1.0000
VR(81)=	46.000	VR(82)=	1.0000	VR(83)=	.53322	VR(84)=	10.000	VR(85)=	170.00	VR(86)=	10.000
VR(87)=	.00000	VR(88)=	.56097	VR(89)=	3.0522	VR(90)=	.85322	VR(91)=	731.89	VR(92)=	27.868
VR(93)=	717.17	VR(94)=	1.9009	VR(95)=	105.00	VR(96)=	.00000	VR(97)=	.00000	VR(98)=	.00000
VR(99)=	.00000	VR(100)=	.00000	VR(							

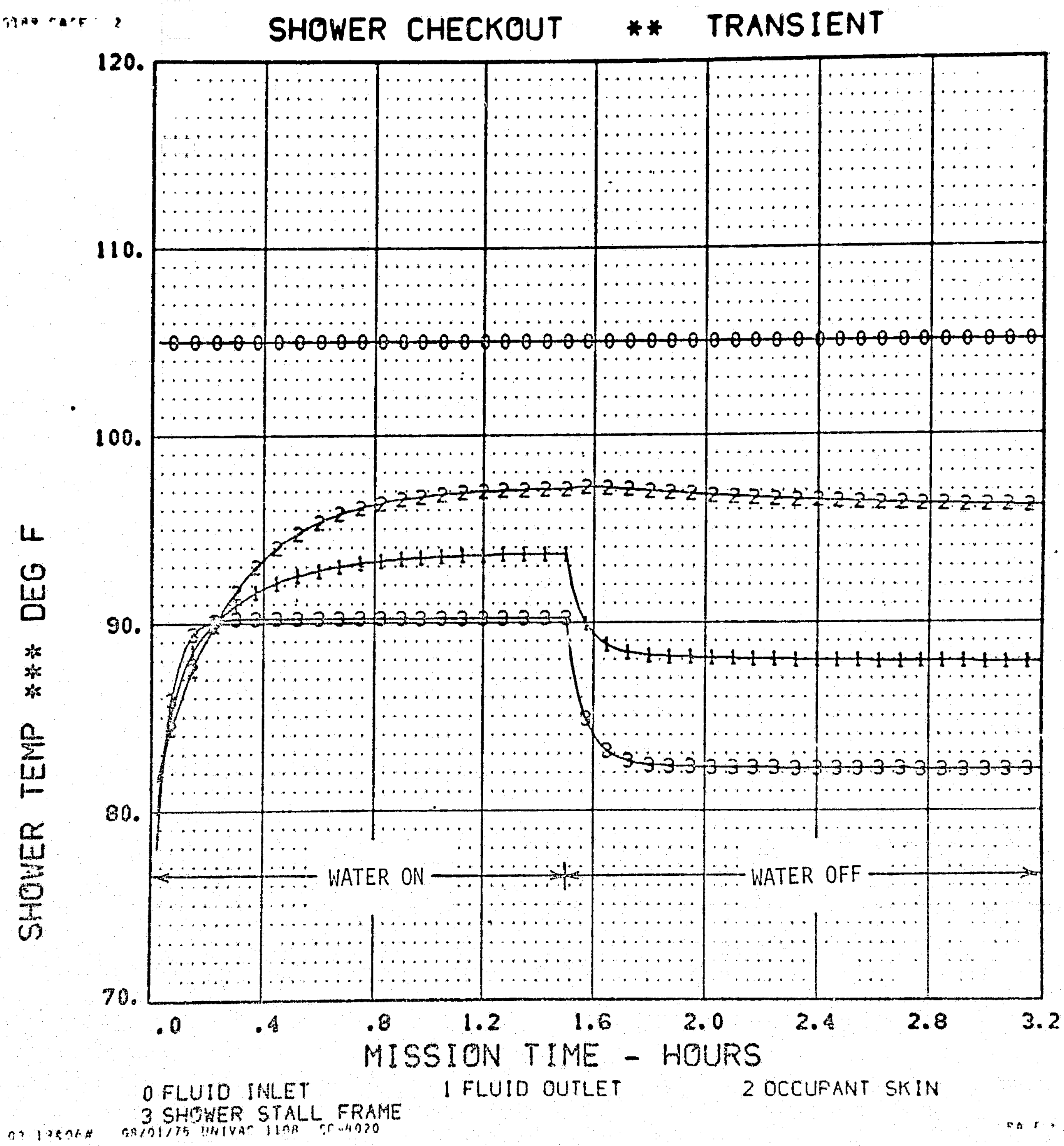

Figure 4-31. Temperatures for Shower Transient Checkout Run

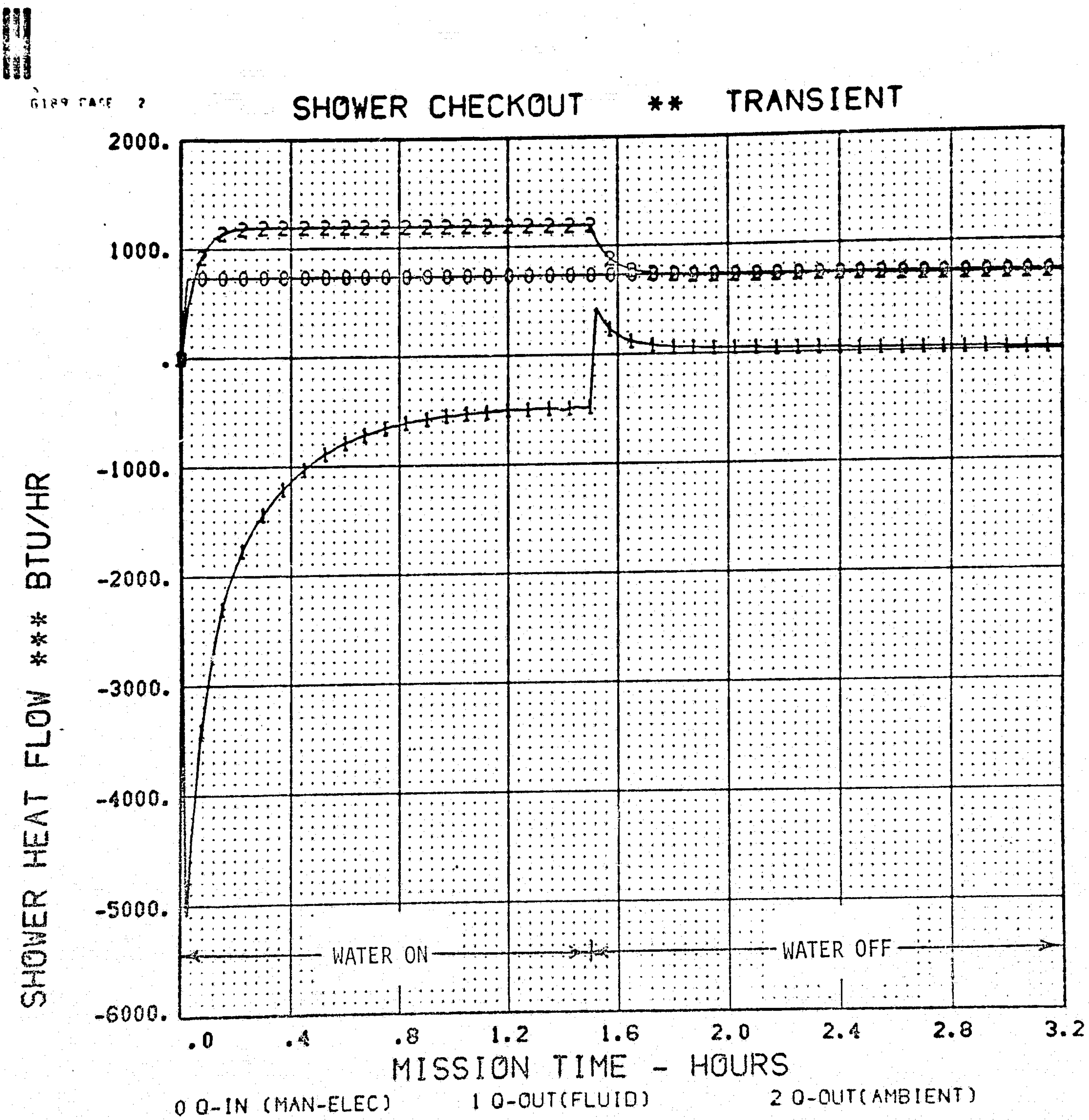

Figure 4-32. Heat Flow for Shower Transient Checkout Run

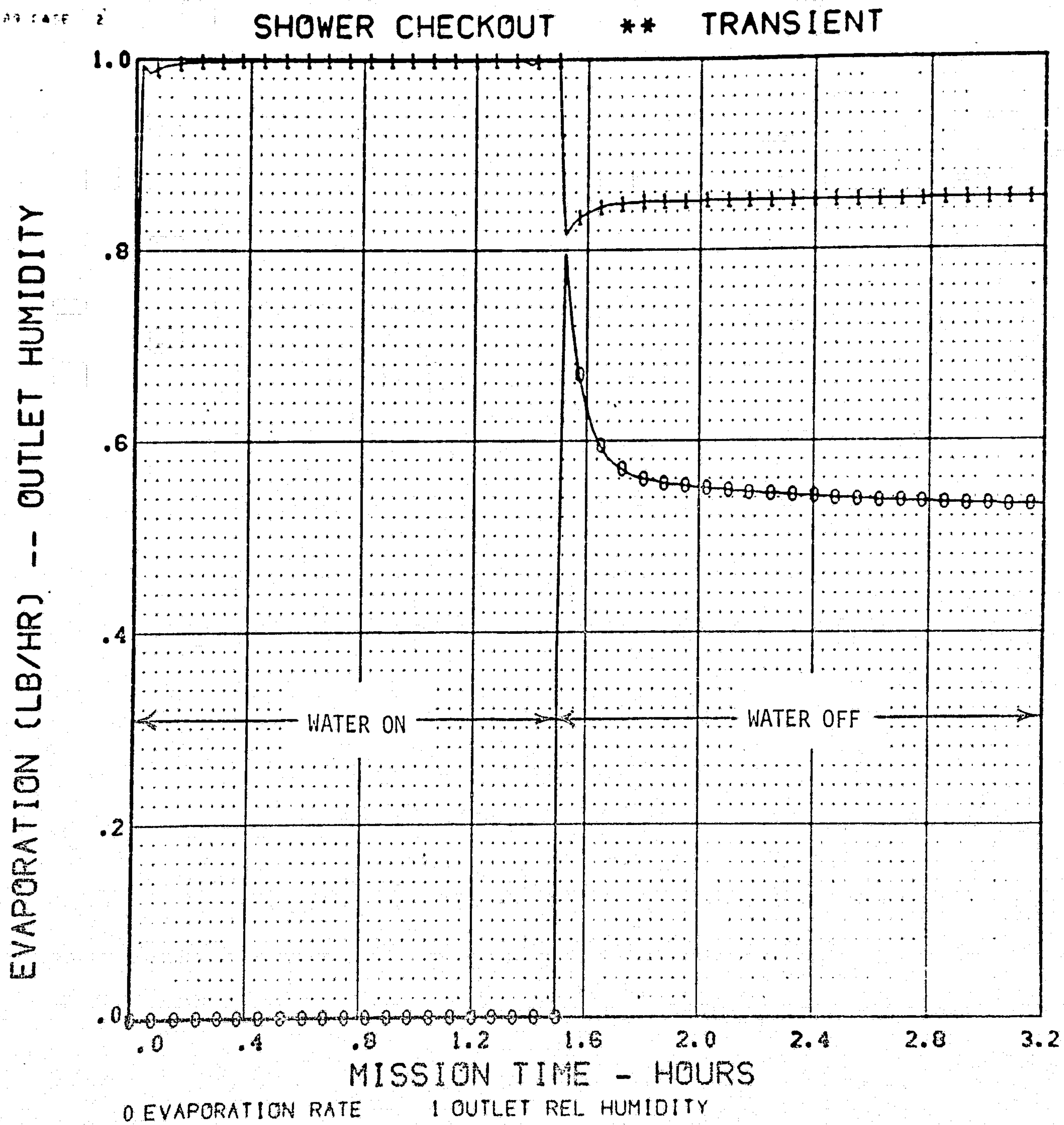

Figure 4-33. Evaporation Parameters for Shower Transient Checkout Run

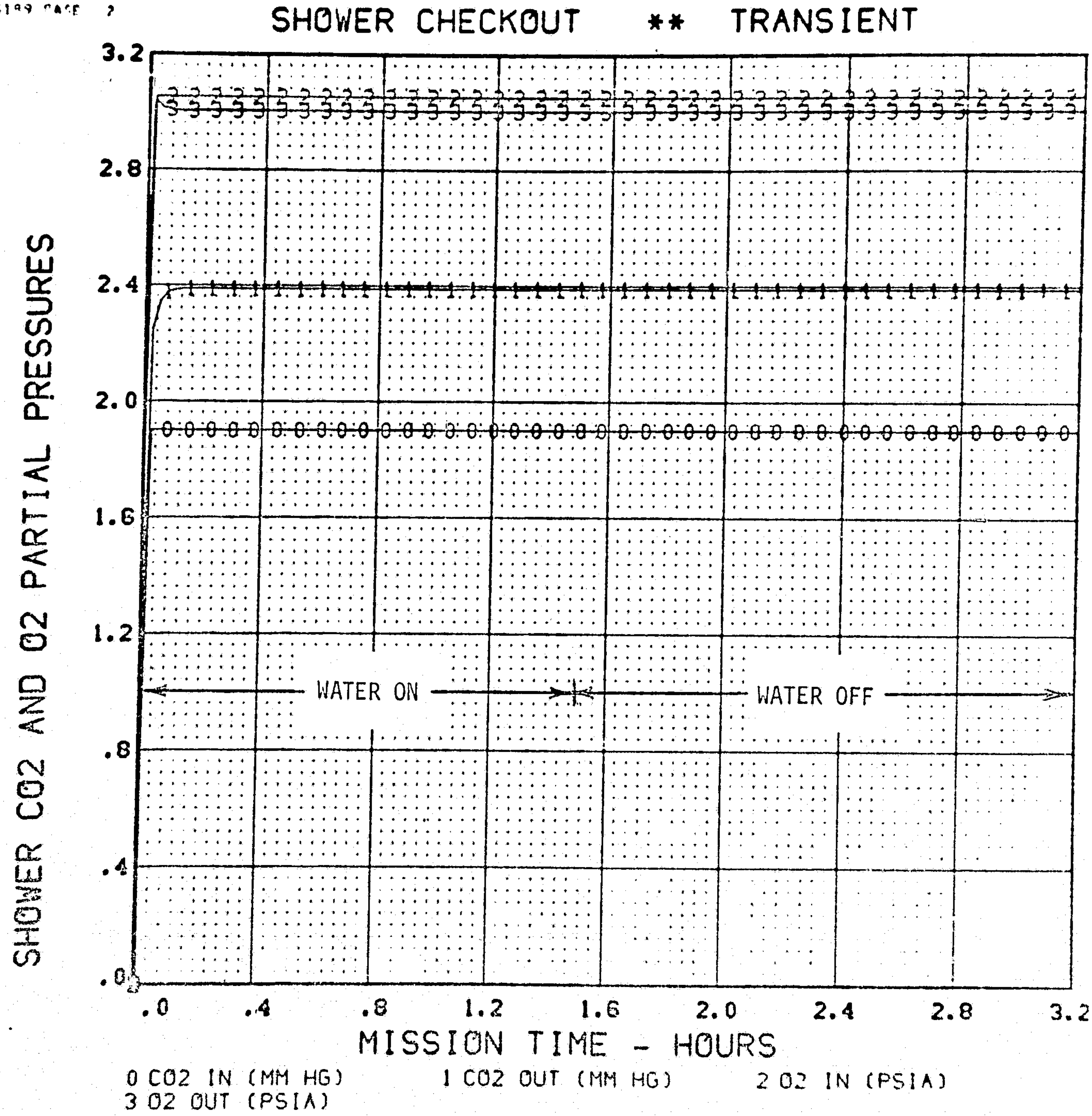

Figure 4-34. CO_2 and O_2 Levels for Shower Transient Checkout Run

4.5 (Continued)

using standard G-189A subroutines already operational. The complete washer/dryer subsystem is included in the Space Station model described in Section 6.

To minimize the high power consumption of an electrical clothes dryer, it is recommended in Reference 21 that a drying time of 4 hours be allowed. This will permit much smaller air flow rates, and possibly lower inlet temperatures, than are used in commercial dryers. Also, a much faster spin-dry speed could be used to remove more of the water before drying. Typical household clothes washers achieve a 60 to 100 percent water retention (lb water/lb dry clothes) in the clothes (Reference 25). However, a high-speed spin is reported in Reference 26 to achieve an 11 percent water retention. Therefore, commercial dryer data could not be used to correlate the dryer model. The model input data were made to correspond to the selected Space Station washer/dryer in Reference 1, and they also agree with the data from References 21 and 27. Since these represent conceptual designs only, and not current hardware, it was necessary to estimate reasonable values for some input data (such as thermal conductors within the washer frame) which will depend on the final design details.

Steady-state and transient checkout runs were made. Results were obtained for four cycle phases: (1) wash water fill; (2) wash circulate; (3) spin-dry, wash water out; and (7) dry. The rinse cycles 4, 5, and 6 are computationally equivalent to the wash cycles and hence were not considered here. They are included, however, in the all-up Space Station model in Section 6.

4.5.1 Clothes Washer/Dryer Steady-State Run

Both the steady-state and transient clothes washer/dryer runs used the subroutine input default data listed in Table 3-16. The final steady-state solution is tabulated in Table 4-15. The steady-state results are plotted in Figures 4-35 through 4-39. These figures show the solution with the unit turned

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TABLE 4-15

FINAL STEADY-STATE SOLUTION FOR CLOTHES WASHER/DRYER CHECKOUT RUN

A(1)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPA=	.00000	WTMA=	.00000	RHOA=	.00000	VISCA=	.00000	XKA=	.00000
B(1)=	12.000	150.00	14.700	14.700	11.906	.93600-01	.00000	.24122	28.902	2.7600
	9.0738	.72600-01	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPB=	.24277	WTMB=	28.766	RHOB=	.64609-01	VISCB=	.47000-01	XKB=	.15400-01

VR(1)=	.00000	VR(2)=	88.482	VR(3)=	.00000	VR(4)=	.00000	VR(20)=	12.185	VR(21)=	88.482
VR(22)=	14.700	VR(23)=	14.700	VR(24)=	11.906	VR(25)=	.27854	VR(26)=	.00000	VR(27)=	.24122
VR(28)=	28.902	VR(29)=	2.7600	VR(30)=	9.0738	VR(31)=	.72600-01	VR(32)=	.00000	VR(33)=	.00000
VR(51)=	89.574	VR(52)=	8.0763	VR(53)=	158.11	VR(54)=	70.000	VR(55)=	3.0000	VR(56)=	58.732
VR(57)=	70.000	VR(58)=	1.0000	VR(59)=	21.071	VR(60)=	70.000	VR(61)=	4.0000	VR(62)=	78.310
VR(63)=	89.577	VR(64)=	.00000	VR(65)=	170.60	VR(66)=	4.0000	VR(67)=	.40000	VR(68)=	50.000
VR(69)=	.00000	VR(70)=	150.00	VR(71)=	.00000	VR(72)=	88.482	VR(73)=	.00000	VR(74)=	.00000
VR(75)=	.20000	VR(76)=	12.000	VR(77)=	.25000	VR(78)=	.50000-01	VR(79)=	240.00	VR(80)=	3343.2
VR(81)=	8.0000	VR(82)=	16.000	VR(83)=	120.00	VR(84)=	.95000	VR(85)=	.00000	VR(86)=	.80000
VR(87)=	10.000	VR(88)=	1.0000	VR(89)=	.40000	VR(90)=	.00000	VR(91)=	.53217	VR(92)=	.34987
VR(93)=	.18494	VR(94)=	.19740-01	VR(95)=	.00000	VR(96)=	.00000	VR(97)=	30.000	VR(98)=	30.000
VR(99)=	.25000	VR(100)=	.00000	VR(101)=	.00000	VR(102)=	55.000	VR(103)=	55.000	VR(104)=	.00000
VR(105)=	10.000	VR(106)=	4.0000	VR(107)=	10.000	VR(108)=	.00000	VR(109)=	.20000	VR(110)=	6.0000
VR(111)=	24.386	VR(112)=	150.00	VR(113)=	.00000	VR(114)=	.00000	VR(115)=	12.000	VR(116)=	.10000
VR(117)=	.18494	VR(118)=	.79522	VR(119)=	7.0000	VR(120)=	89.000	VR(121)=	.00000	VR(122)=	.00000

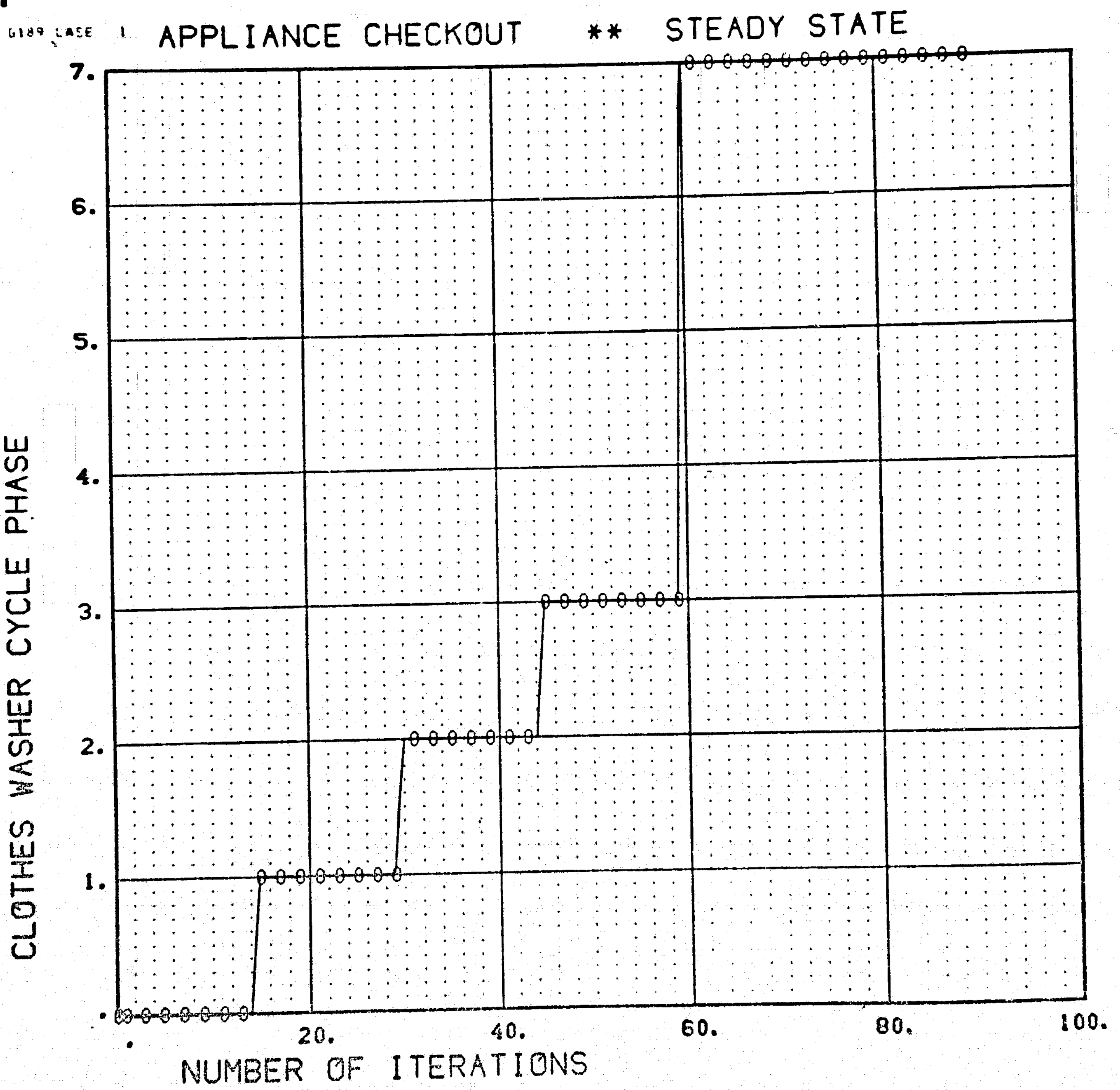

Figure 4-35. Operational Cycle Phase for Clothes Washer/Dryer Steady-State Checkout Run

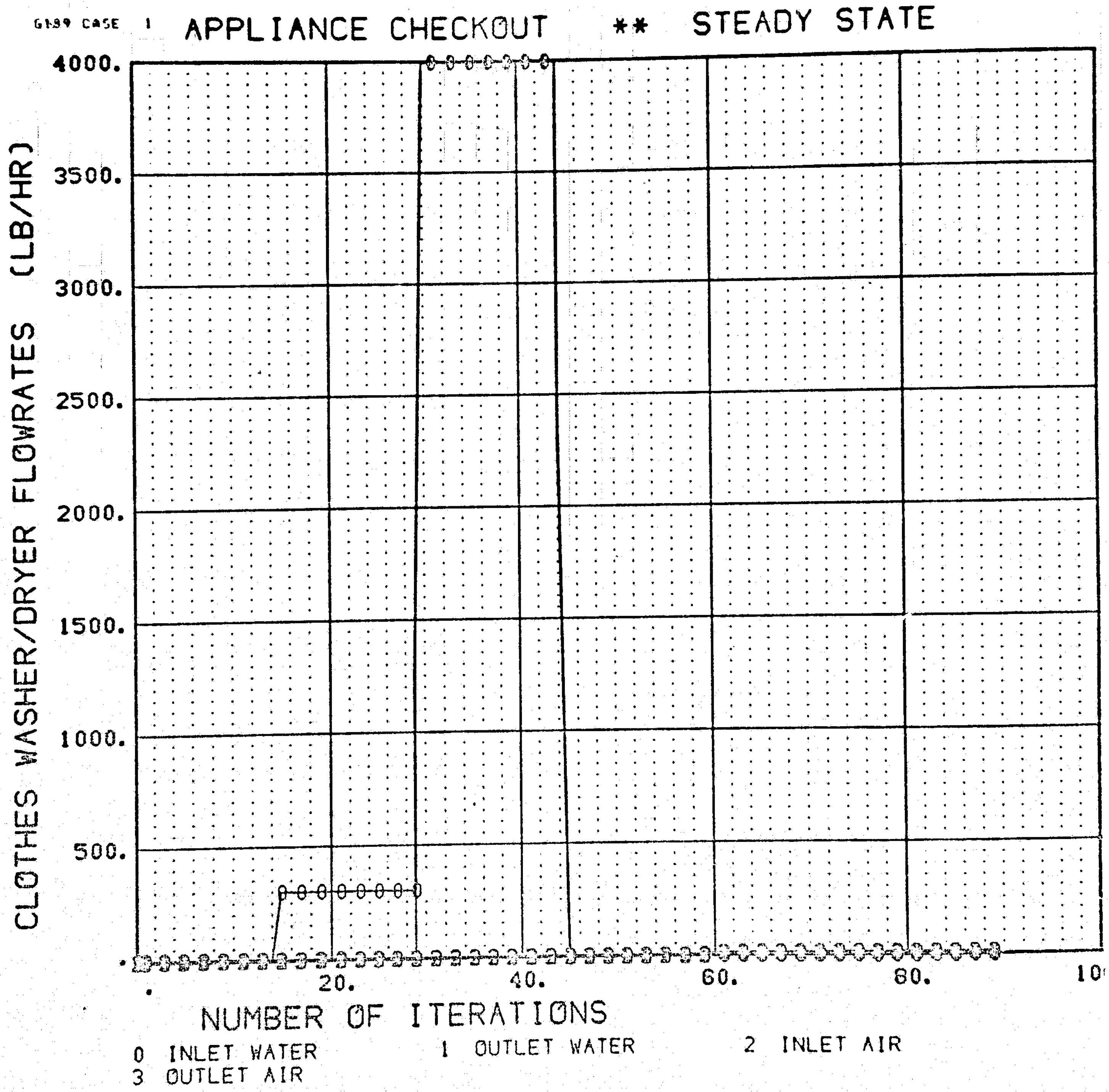

Figure 4-36. Flow Rates for Clothes Washer/Dryer Steady-State Checkout Run

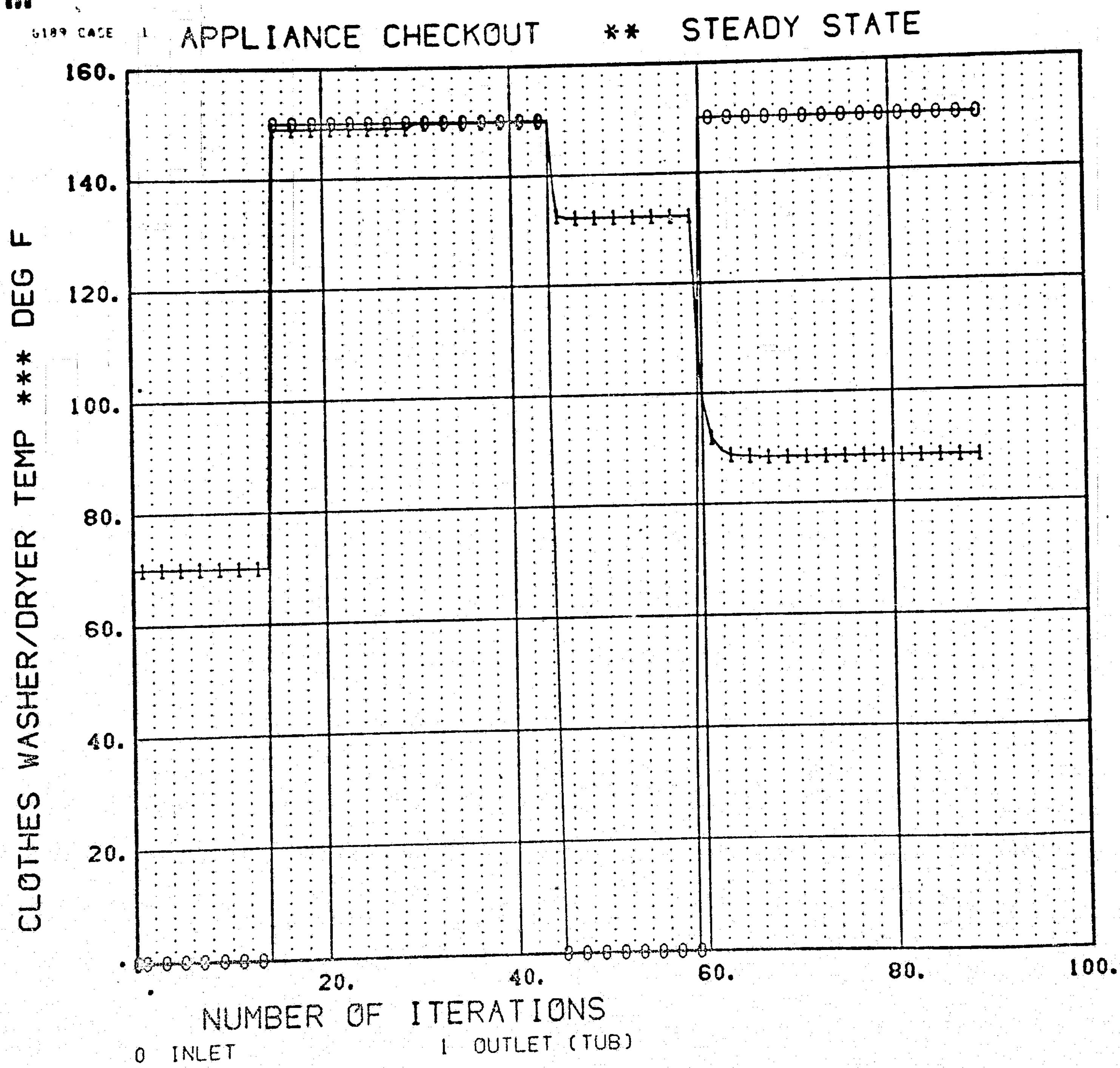

Figure 4-37. Temperatures for Clothes Washer/Dryer Steady-State Checkout Run

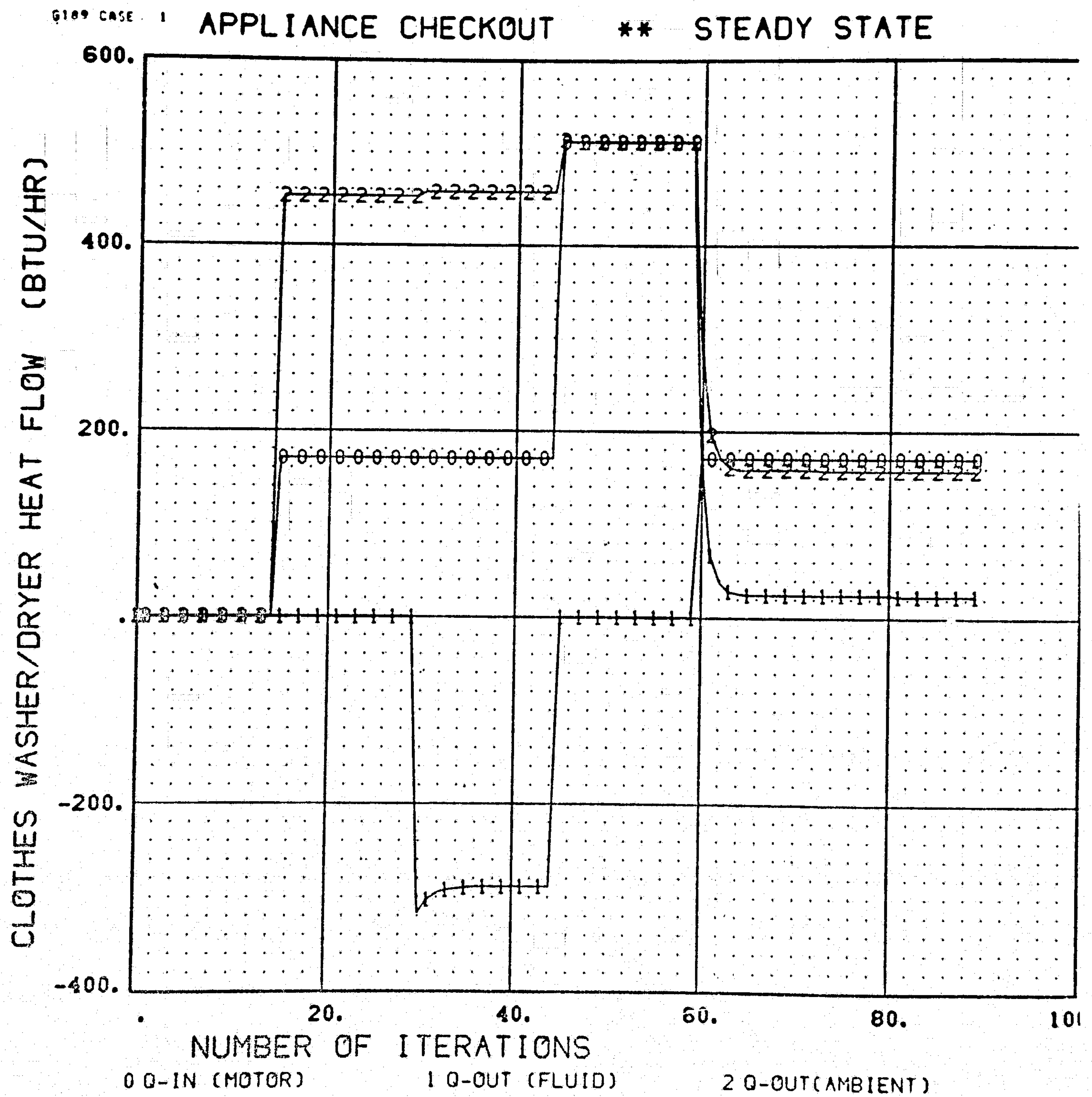

Figure 4-38. Heat Flow for Clothes Washer/Dryer Steady-State Checkout Run

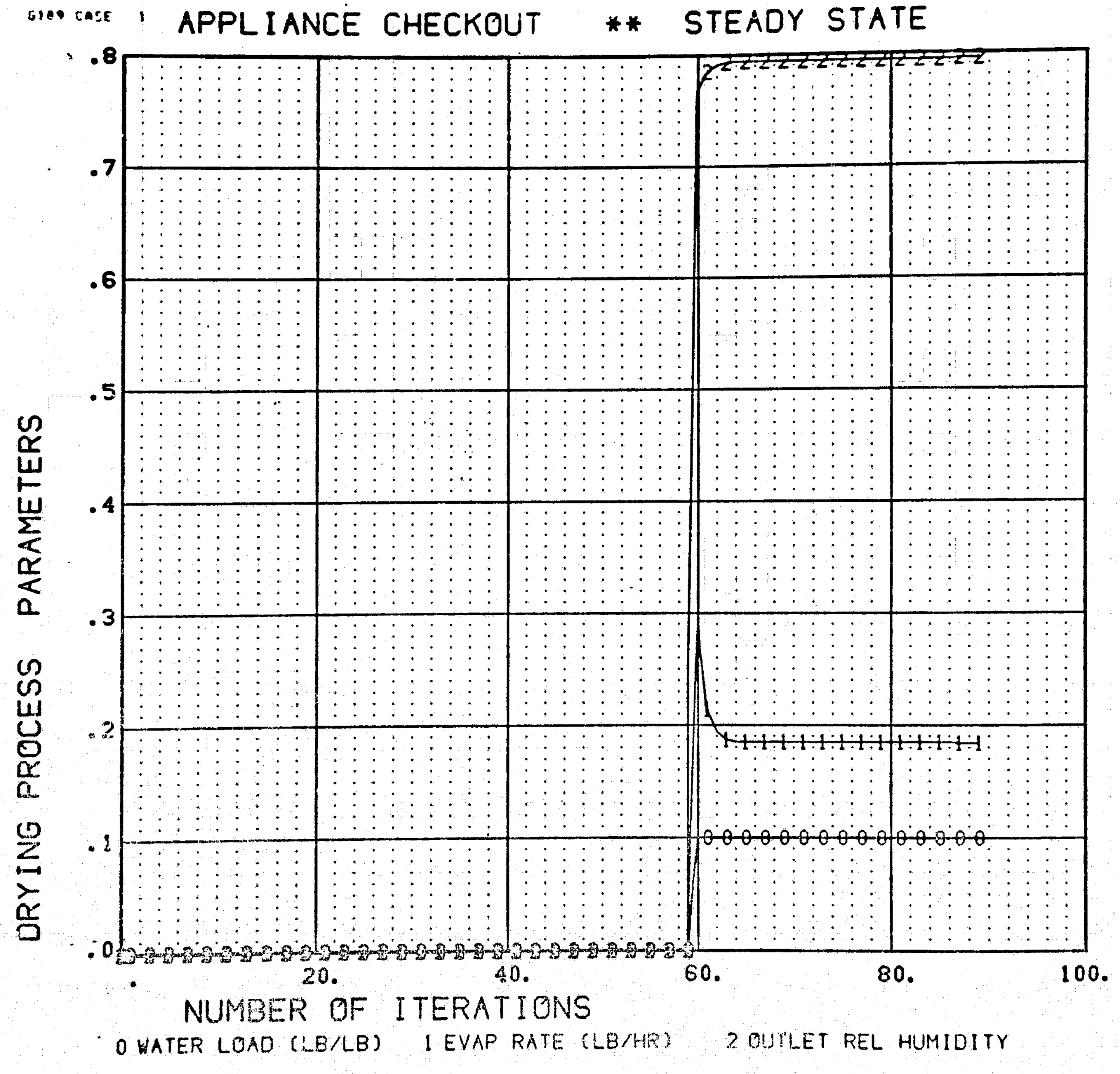

Figure 4-39. Evaporation Parameters for Clothes Washer/Dryer Steady-State Checkout Run

4.5.1 (Continued)

off and for cycle phase 1 (wash water fill), 2 (wash-circulate), 3 (spin-dry), and 7 (dry). Since the solution was steady state, rather than transient, the amount of water in the drum was held constant, even in cases (such as wash water fill) where there was a net flow of water into or out of the drum.

During the initial phase with the unit turned off, all fluid flow rates and heat flow in and out of the drum are seen in Figures 4-36 and 4-38 to be zero; and the washer temperature was equal to the assumed ambient temperature of 70°F as required. During phase 1, wash water fill, the heat input from the assumed 50 watt drum motor was 170.6 Btu/hr. The net heat conducted through the structure to ambient is shown in Figure 4-38 to be 450 Btu/hr. Thus, the total heat input from the inlet fluid must have been 450-170.6 = 279.4 Btu/hr. For the inlet water flow rate of 300 lb/hr, its energy contribution may also be expressed as:

$$q_{\text{input from water}} = \dot{m}\, c_p\, \Delta T$$

$$279.4 \text{ Btu/hr} = (300 \text{ lb/hr})(1 \text{ Btu/lb °F})(150\text{°F} - T_{final}) \tag{4.5.1}$$

Solving the above equation results in a required final water (and tub) temperature of 149°F. This is the actual predicted temperature shown in Figure 4-37, thus verifying the accuracy of the energy balance.

For the steady-state run, the next cycle phase 2 (wash-circulate) differs from phase 1 only in the inlet water flow rate which was 4000 lb/hr as shown in Figure 4-36. This flow rate was recommended in Reference 21 to provide adequate agitation for washing. Substituting this flow rate into equation (4.5.1) results in a final water and tub temperature of approximately 150°F. This value is reflected in Figure 4-37.

4.5.1 (Continued)

During phase 3 (spin-dry), the only heat input to the washer drum is from the 150 watt motor or 511.8 Btu/hr. Since all this heat is applied to the frame node in the thermal model, the tub and frame should be at the same temperature recording to the following relation:

$$q_{MOTOR} = G_{frame\text{-}to\text{-}ambient} (T_{tub} - T_{ambient})$$

$$511.8 \text{ Btu/hr} = 8.216 \text{ Btu/hr-°F} (T_{tub} - 70°F)$$

Solving this equation results in a washer tub and water temperature of 132.2°F, which is also the value reflected in Figure 4-37.

The final steady-state solution was for the drying phase 7. There was assumed to be 0.4 lb of water [R(67)] in the clothes load of 4 lb [R(66)]. This gave a water loading of 10 percent as shown in Figure 4-39. For this condition, and with 12 lb/hr inlet air flow rate, the predicted evaporation rate was 0.185 lb/hr in Figure 4-39. Since typical household clothes dryers use much higher air flow rates, these data cannot be correlated with conventional dryer tests. If test data become available in the future for a spacecraft dryer, correlation of the model with the data may be made easily using the evaporation rate correlation multiplier, R(89), described in Paragraph 3.5.3.2. For the 0.185 lb/hr evaporation rate predicted here, the latent cooling effect would be 193 Btu/hr. This is nearly balanced by the drum motor input power of 50 watts or 171 Btu/hr. In addition, the heat conducted through the dryer structure and dissipated to ambient is given by

$$q_{structure\ loss} = G\,\Delta T = (8.076 \text{ Btu/hr °F})(89.6°F - 70°F)$$

$$= 158.3 \text{ Btu/hr}$$

4.5.1 (Continued)

Thus, the net heat input required from the air flow should be (193-171+158)= 180 Btu/hr. This heat should be governed by the equation

$$q_{out\ of\ air} = \dot{m}\ c_p\ (T_{in} - T_{out})$$

$$180\ \text{Btu/hr} = (12\ \text{lb/hr})(0.243\ \text{Btu/lb}\ ^\circ\text{F})(150^\circ\text{F} - T_{out})$$

and the final air outlet temperature should thus be 88.3°F. This is indeed the predicted outlet air temperature in Figure 4-37, thus verifying the accuracy of the energy balance in the model.

4.5.2 Clothes Washer/Dryer Transient Run

The G-189A input data for the transient clothes washer/dryer run were identical to the steady-state case. The final transient solution is tabulated in Table 4-16. The transient plots are shown in Figures 4-40 through 4-44. The washer was initially off, then cycled through phase 1 (wash water fill); 2 (wash-circulate); 3 (spin-dry, wash water out); and 7 (dry). The rinse phases were omitted here since they are equivalent to the wash phases. (Note that they are included in the all-up Space Station run in Section 6.)

During the initial phase with the unit turned off, all flow rates and heat flow are zero; and the washer temperatures are equal to the assumed ambient temperature of 70°F. The solution during phase 1 (wash water fill) can be seen to be generally approaching the steady-state solution discussed previously. The solution during phase 2 (wash water circulate) does reach the steady-state solution described previously. The drying phase 7 can be compared at time 2.5 hours with the steady-state case discussed previously since the water loading in the clothes is the same for both. The two solutions are very nearly equal, with the transient temperatures being only slightly higher due to the stored energy in the dryer drum from the previous wash cycle.

TABLE 4-16

FINAL TRANSIENT SOLUTION FOR CLOTHES WASHER/DRYER CHECKOUT RUN

VR(1)=	.00000	VR(2)=	92.434	VR(3)=	.00000	VR(4)=	.00000	VR(20)=	.00000	VR(21)=	92.434
VR(22)=	.00000	VR(23)=	.00000	VR(24)=	.00000	VR(25)=	.00000	VR(26)=	.00000	VR(27)=	.00000
VR(28)=	.00000	VR(29)=	.00000	VR(30)=	.00000	VR(31)=	.00000	VR(32)=	.00000	VR(33)=	.00000
VR(51)=	90.461	VR(52)=	8.0801	VR(53)=	168.31	VR(54)=	70.000	VR(55)=	3.0000	VR(56)=	62.492
VR(57)=	70.000	VR(58)=	1.0000	VR(59)=	22.498	VR(60)=	70.000	VR(61)=	4.0000	VR(62)=	83.322
VR(63)=	90.831	VR(64)=	.00000	VR(65)=	.00000	VR(66)=	4.0000	VR(67)=	30.000	VR(68)=	50.000
VR(69)=	.00000	VR(70)=	150.00	VR(71)=	.00000	VR(72)=	92.434	VR(73)=	.00000	VR(74)=	.00000
VR(75)=	.20000	VR(76)=	12.000	VR(77)=	.25000	VR(78)=	.50000-01	VR(79)=	240.00	VR(80)=	3343.2
VR(81)=	8.0000	VR(82)=	16.000	VR(83)=	120.00	VR(84)=	.95000	VR(85)=	.00000	VR(86)=	.80000
VR(87)=	10.000	VR(88)=	1.0000	VR(89)=	.40000	VR(90)=	.00000	VR(91)=	.43272	VR(92)=	.00000
VR(93)=	.00000	VR(94)=	.19740-01	VR(95)=	.00000	VR(96)=	.19961	VR(97)=	30.000	VR(98)=	30.000
VR(99)=	24.250	VR(100)=	.00000	VR(101)=	.00000	VR(102)=	55.000	VR(103)=	55.000	VR(104)=	.25000-01
VR(105)=	10.000	VR(106)=	4.0000	VR(107)=	10.000	VR(108)=	-1.0000	VR(109)=	.20000	VR(110)=	6.0000
VR(111)=	.00000	VR(112)=	.00000	VR(113)=	.00000	VR(114)=	.00000	VR(115)=	.00000	VR(116)=	.00000
VR(117)=	.00000	VR(118)=	.00000	VR(119)=	.00000	VR(120)=	152.00	VR(121)=	3.8250	VR(122)=	.00000

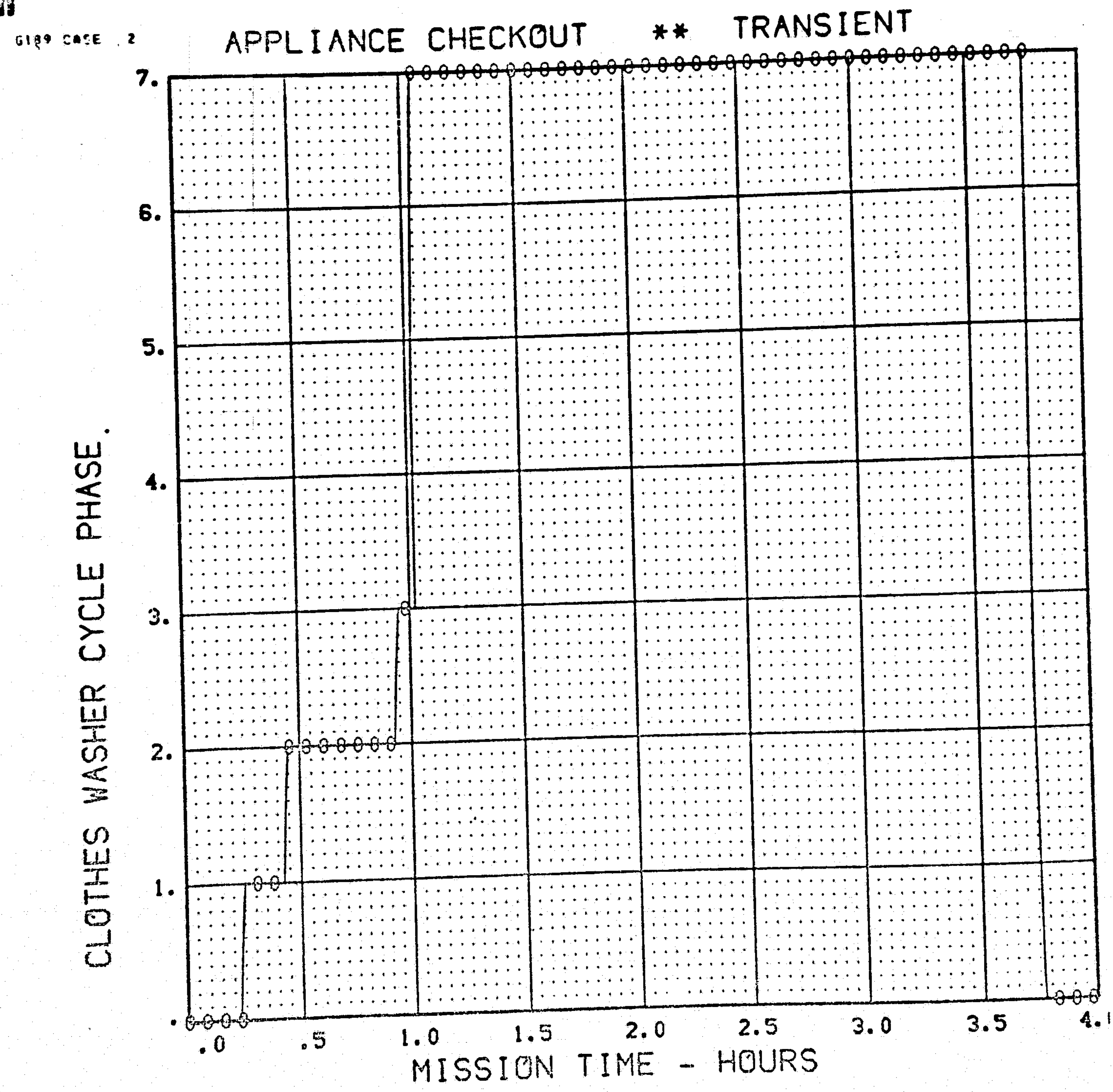

Figure 4-40. Operational Cycle Phase for Clothes Washer/Dryer Transient Checkout Run

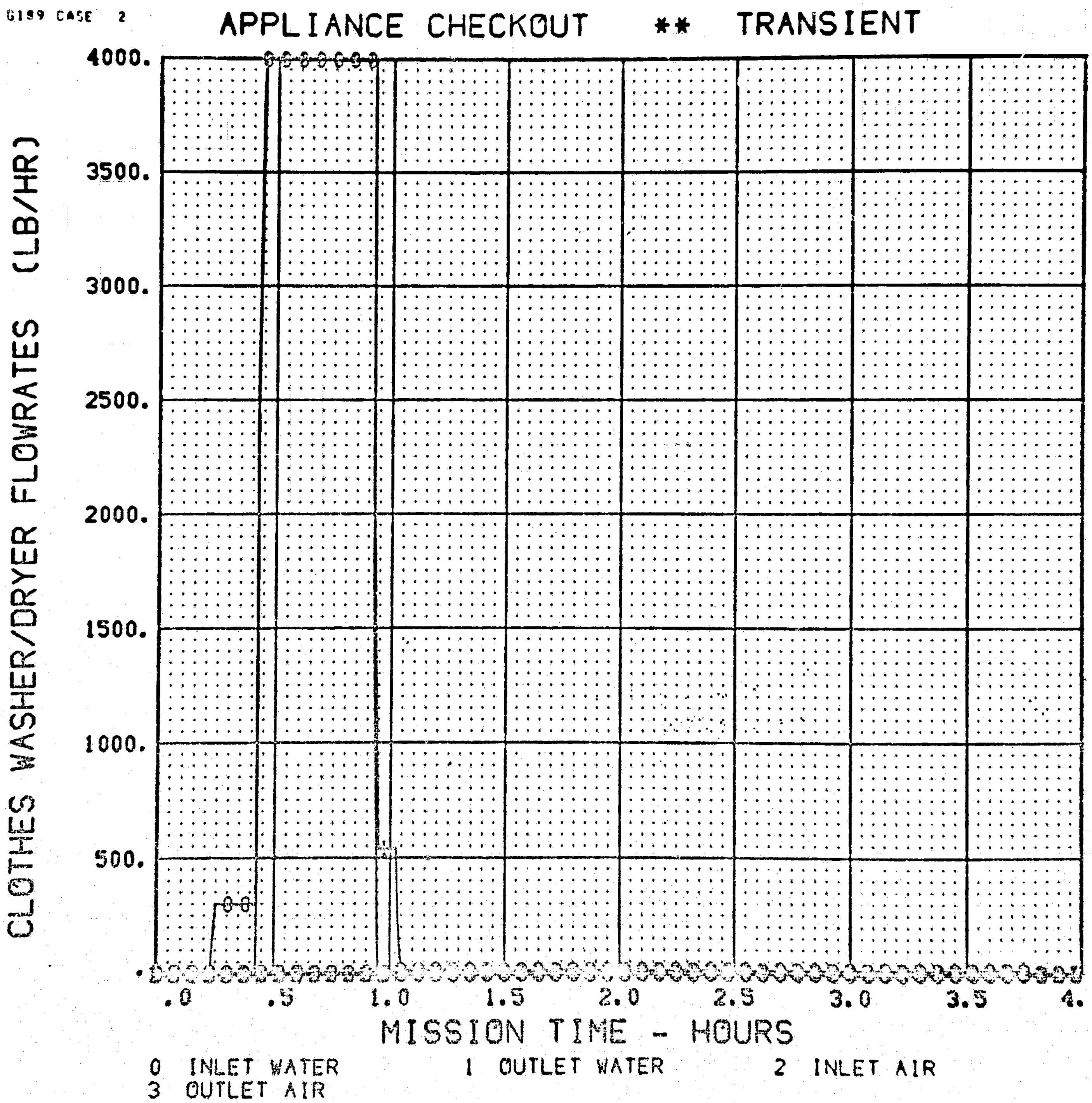

Figure 4-41. Flow Rates for Clothes Washer/Dryer Transient Checkout Run

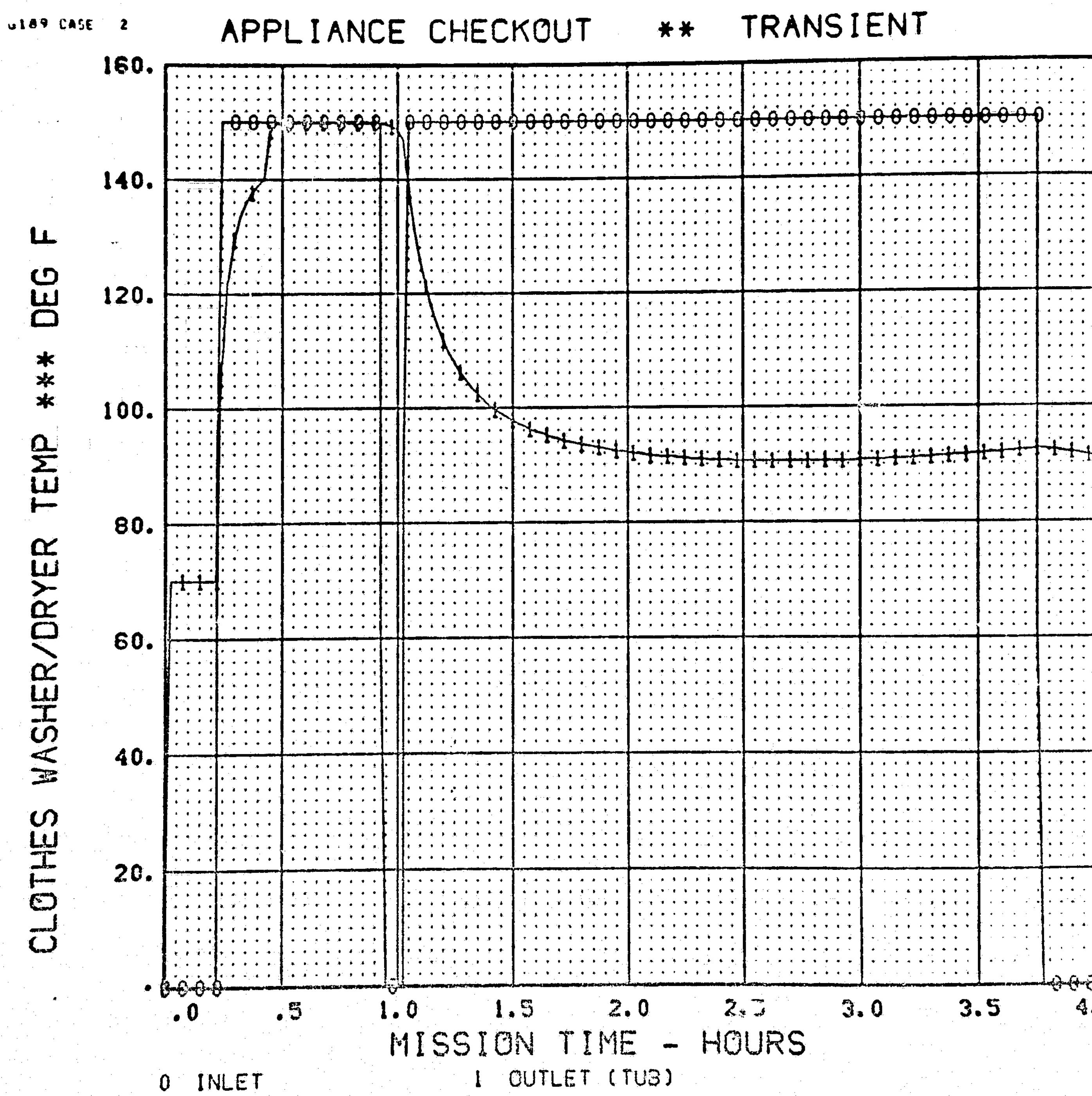

Figure 4-42. Temperatures for Clothes Washer/Dryer Transient Checkout Run

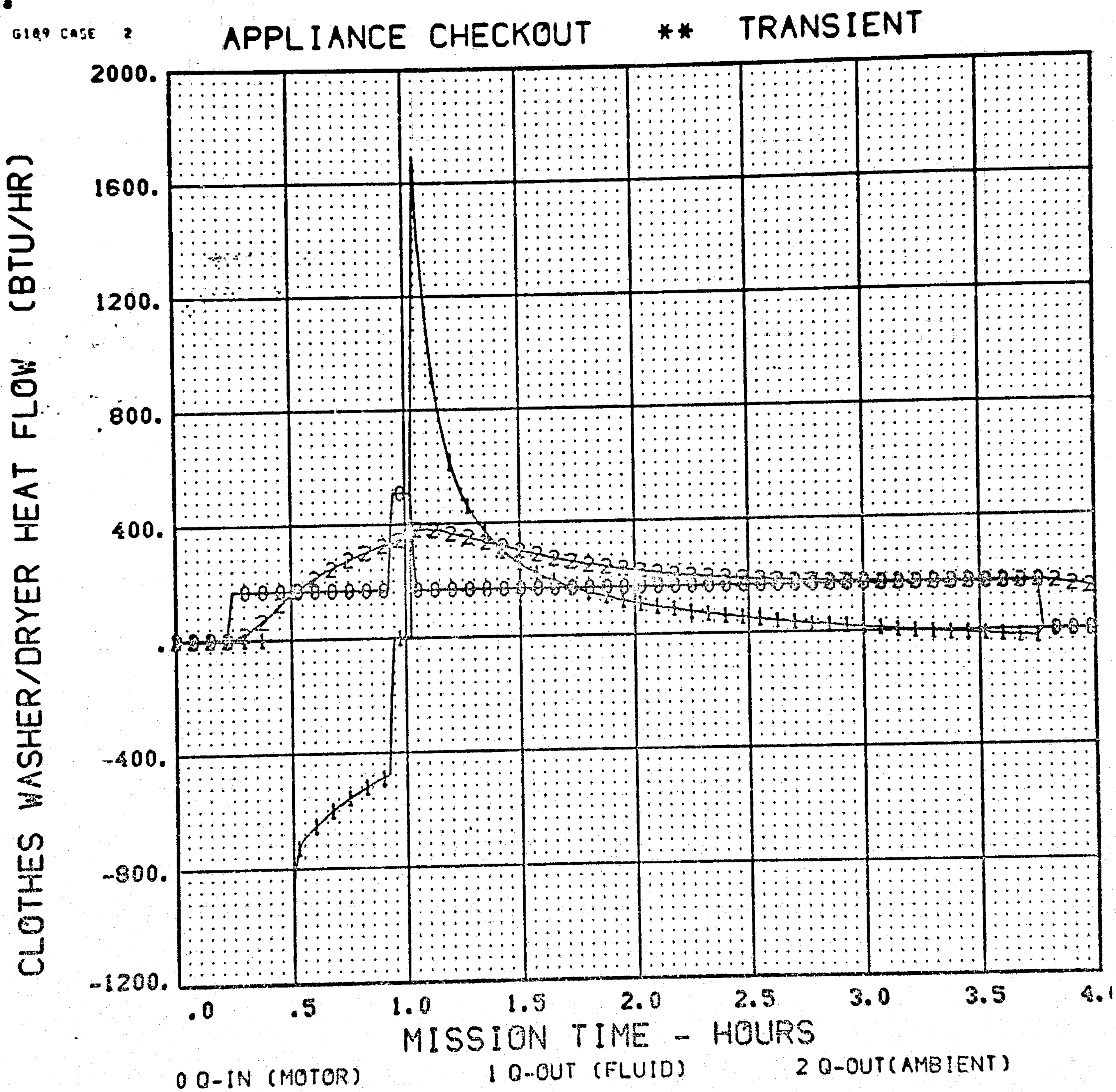

Figure 4-43. Heat Flow for Clothes Washer/Dryer Transient Checkout Run

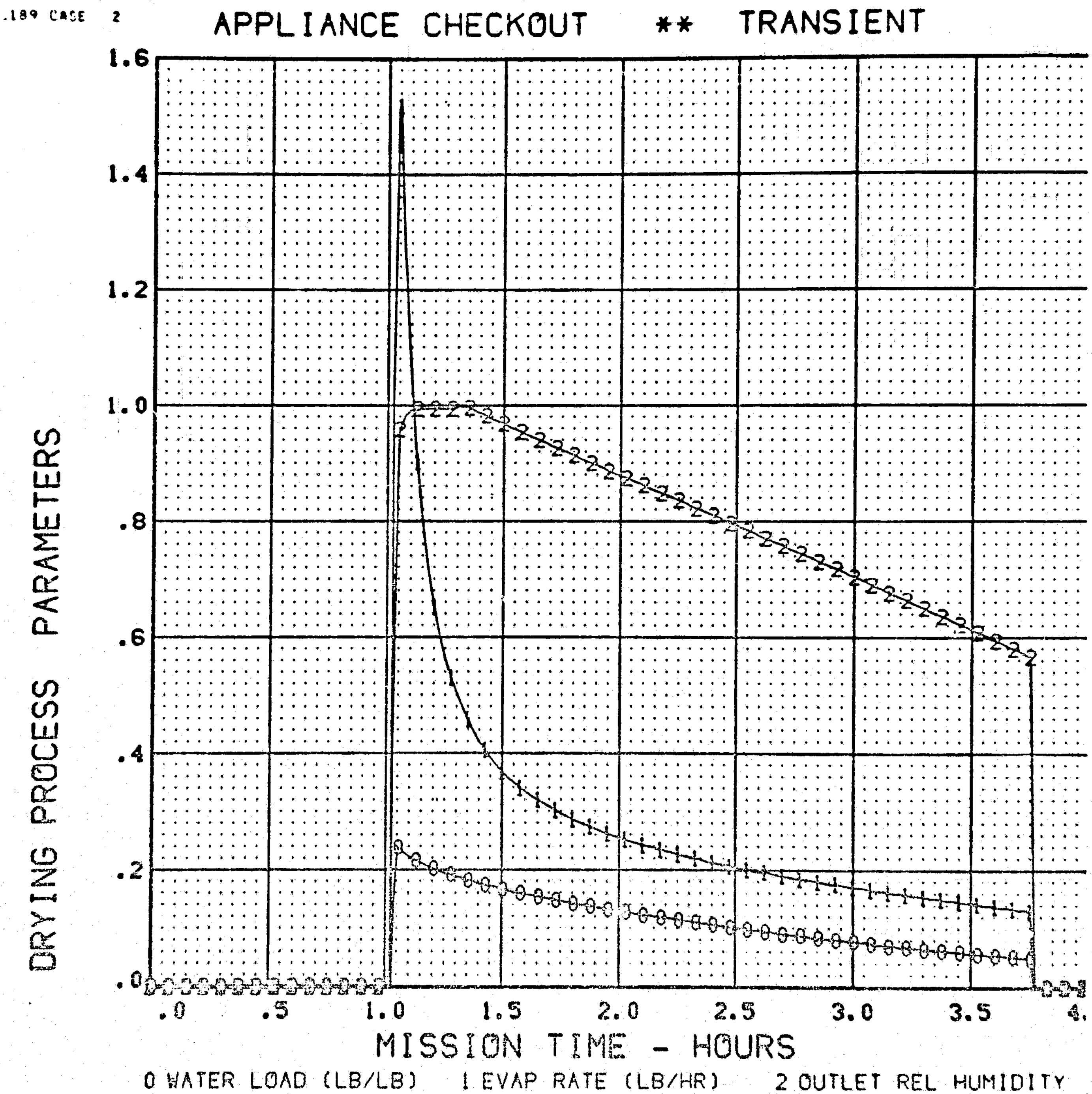

Figure 4-44. Evaporation Parameters for Clothes washer/Dryer Transient Checkout Run

4.5.3 Dishes Washer/Dryer

Both the steady-state and transient dishwasher/dryer runs used the subroutine input default data listed in Table 3-16. The final steady-state solution is tabulated in Table 4-17. The steady-state results are plotted in Figures 4-45 through 4-49. The transient solution is shown in Table 4-18 and Figures 4-50 through 4-54. These results were checked in the same manner as was done for the clothes washer/dryer case and found to be valid.

4.6 WASTEC CHECKOUT RUN

The WASTEC subroutine was checked out by modeling the dryjohn planned for use on Shuttle. Only the collector and urinal were modeled for the checkout runs since the peripheral equipment (pump, valves, lines, etc.) can be modeled using standard G-189A subroutines already operational. The complete dryjohn subsystem is included in the Shuttle and Space Station models in Sections 5 and 6.

A steady-state and transient run were made as described in the following paragraphs. The subroutine input default data, listed in Table 3-19, were used for both cases, except for two minor exceptions. First, the initial collector temperature in R(51) was set at 70°F (instead of the 60°F default value). Secondly, the vacuum evaporation correlation multiplier, R(75), was input at 2.5 (instead of the 3.0 default value). The reason for this slight difference is due to subsequent data analysis which showed the default values to be slightly more realistic. The default values were used in the Space Station models in Section 6.

The three possible usage phases described in Paragraph 3.6 were simulated: (1) urine collection; (2) fecal collection; and (3) combined urine/fecal collection, as well as phase 0 (unit off). During phases 0 and 1, the collector contents were assumed to be under vacuum.

4.6.1 Dryjohn Steady-State Run

The final steady-state solution is tabulated in Table 4-19. The steady-state results for the four usage phases are plotted in Figures 4-55 through 4-58. In each case, steady-state conditions were achieved in about five iterations or less.

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TABLE 4-17

FINAL STEADY-STATE SOLUTION FOR DISHWASHER/DRYER CHECKOUT RUN

A(1)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPA=	.00000	WTMA=	.00000	RHOA=	.00000	VISCA=	.00000	XKA=	.00000
B(1)=	38.000	150.00	14.700	14.700	37.704	.29640	.00000	.24122	28.902	8.7400
	28.734	.22990	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPB=	.24277	WTMB=	28.766	RHOB=	.64609-01	VISCB=	.47000-01	XKB=	.15400-01

VR(1)=	.00000	VR(2)=	102.43	VR(3)=	.00000	VR(4)=	.00000	VR(20)=	38.369	VR(21)=	102.43
VR(22)=	14.700	VR(23)=	14.700	VR(24)=	37.704	VR(25)=	.66546	VR(26)=	.00000	VR(27)=	.24122
VR(28)=	28.902	VR(29)=	8.7400	VR(30)=	28.734	VR(31)=	.22990	VR(32)=	.00000	VR(33)=	.00000
VR(51)=	97.870	VR(52)=	8.1016	VR(53)=	225.81	VR(54)=	70.000	VR(55)=	3.0000	VR(56)=	83.617
VR(57)=	70.000	VR(58)=	1.0000	VR(59)=	30.705	VR(60)=	70.000	VR(61)=	4.0000	VR(62)=	111.49
VR(63)=	97.872	VR(64)=	.00000	VR(65)=	170.60	VR(66)=	15.200	VR(67)=	.30000	VR(68)=	50.000
VR(69)=	.00000	VR(70)=	150.00	VR(71)=	.00000	VR(72)=	102.43	VR(73)=	.00000	VR(74)=	.00000
VR(75)=	.25000	VR(76)=	12.000	VR(77)=	.25000	VR(78)=	.10000-03	VR(79)=	240.00	VR(80)=	10587.
VR(81)=	8.0000	VR(82)=	16.000	VR(83)=	120.00	VR(84)=	.95000	VR(85)=	.00000	VR(86)=	.80000
VR(87)=	.45000	VR(88)=	1.0000	VR(89)=	.40000	VR(90)=	.00000	VR(91)=	.40511	VR(92)=	.36906
VR(93)=	.36906	VR(94)=	.19740-01	VR(95)=	.00000	VR(96)=	.00000	VR(97)=	30.000	VR(98)=	30.000
VR(99)=	.25000	VR(100)=	.00000	VR(101)=	.00000	VR(102)=	15.000	VR(103)=	15.000	VR(104)=	.00000
VR(105)=	10.000	VR(106)=	4.0000	VR(107)=	10.000	VR(108)=	.00000	VR(109)=	.10000-01	VR(110)=	6.0000
VR(111)=	-30.314	VR(112)=	150.00	VR(113)=	.00000	VR(114)=	.00000	VR(115)=	38.000	VR(116)=	.30000
VR(117)=	.36906	VR(118)=	.39566	VR(119)=	7.0000	VR(120)=	.00000	VR(121)=	.00000	VR(122)=	.00000

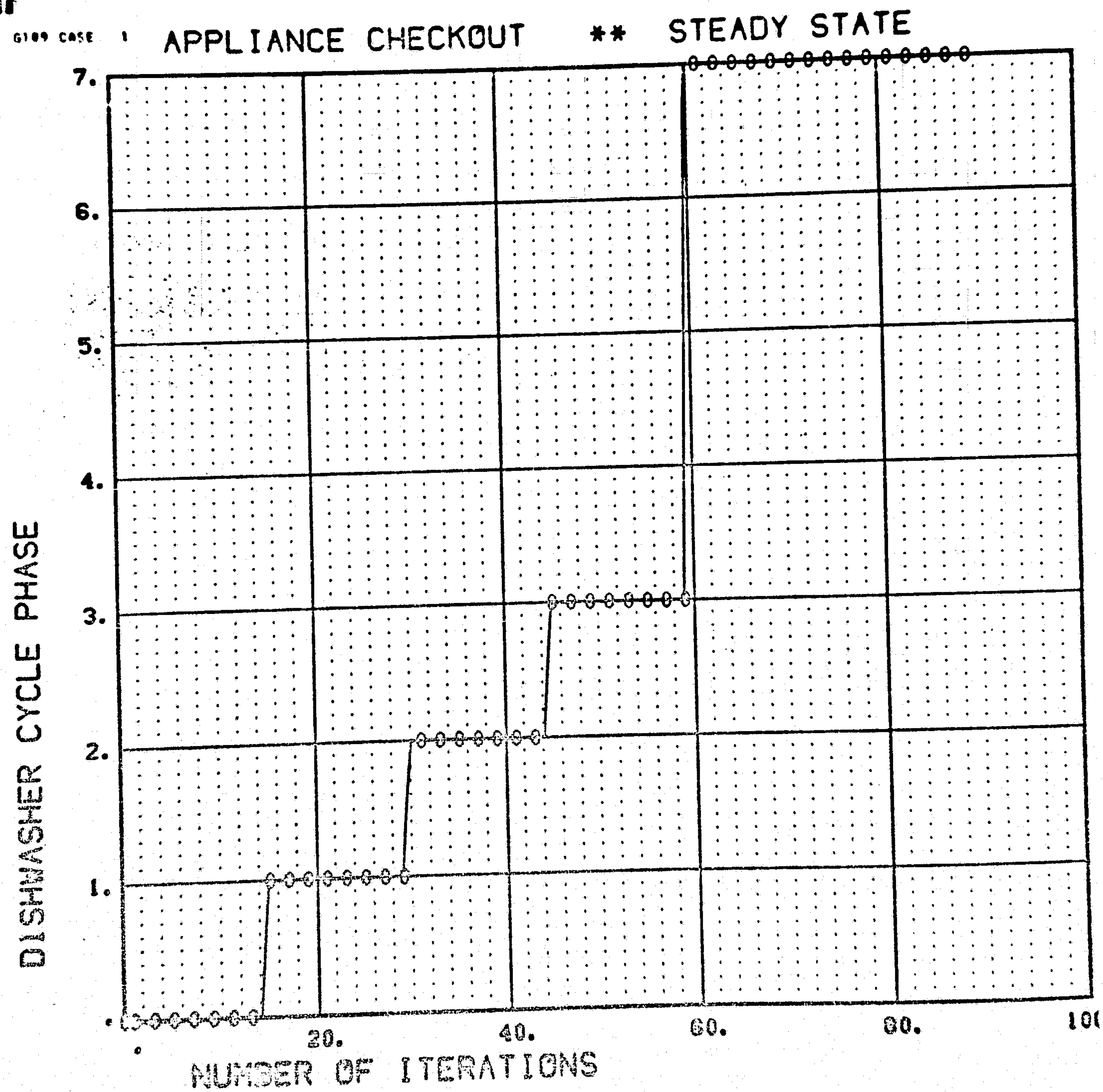

Figure 4-45. Operational Cycle Phase for Dishwasher/Dryer Steady-State Checkout Run

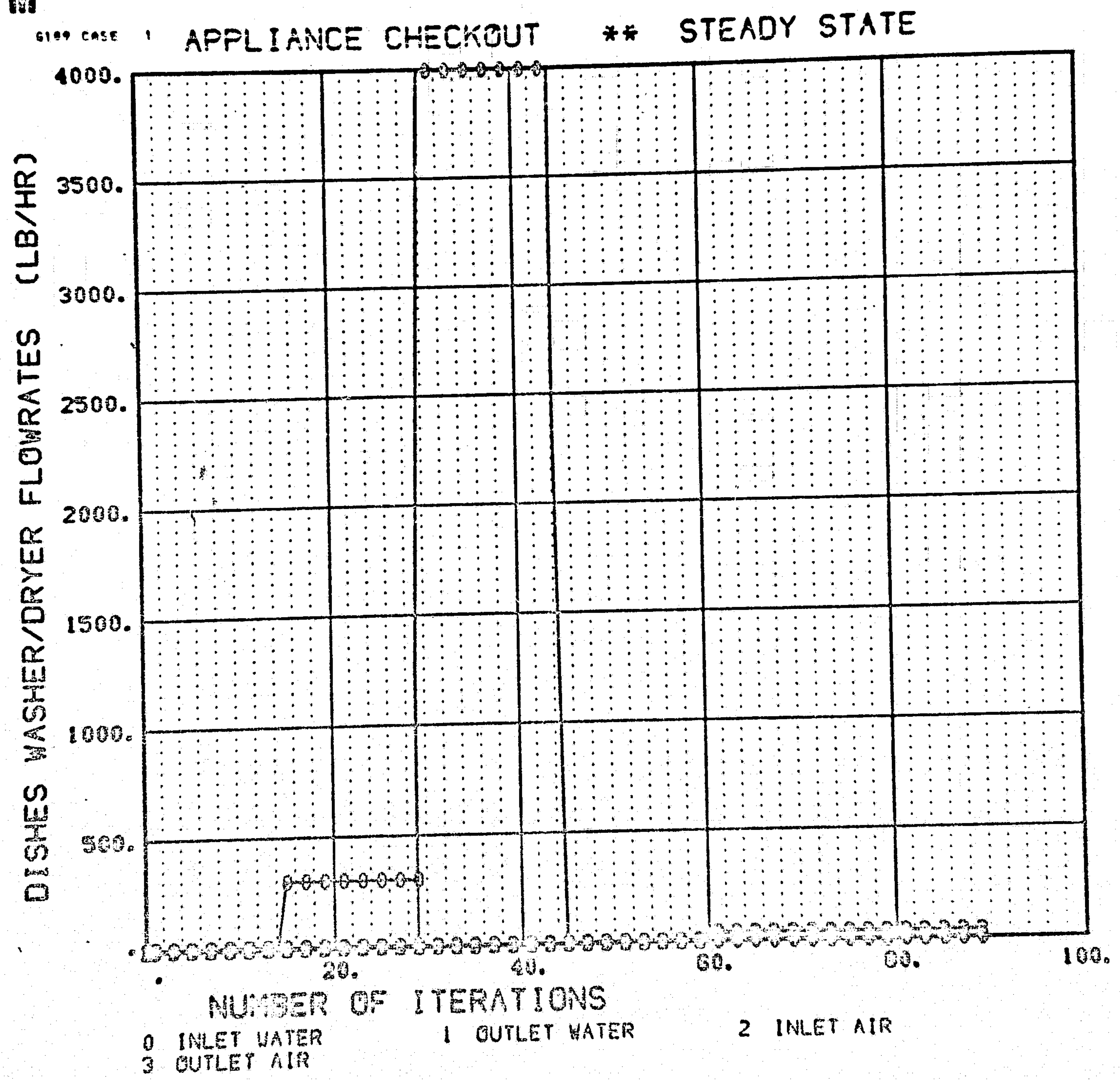

Figure 4-46. Flow Rates for Dishwasher/Dryer Steady-State Checkout Run

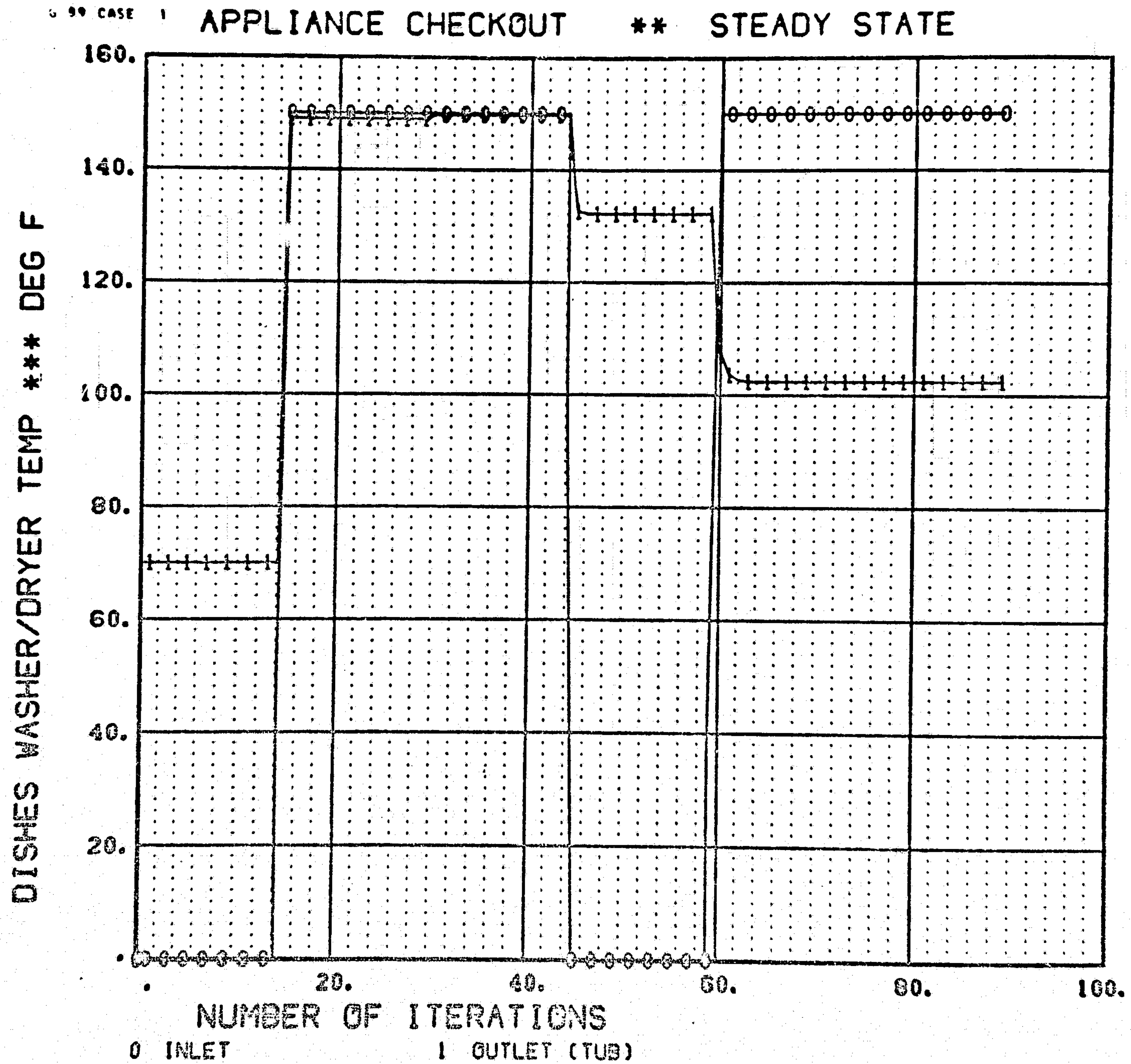

Figure 4-47. Temperatures for Dishwasher/Dryer Steady-State Checkout Run

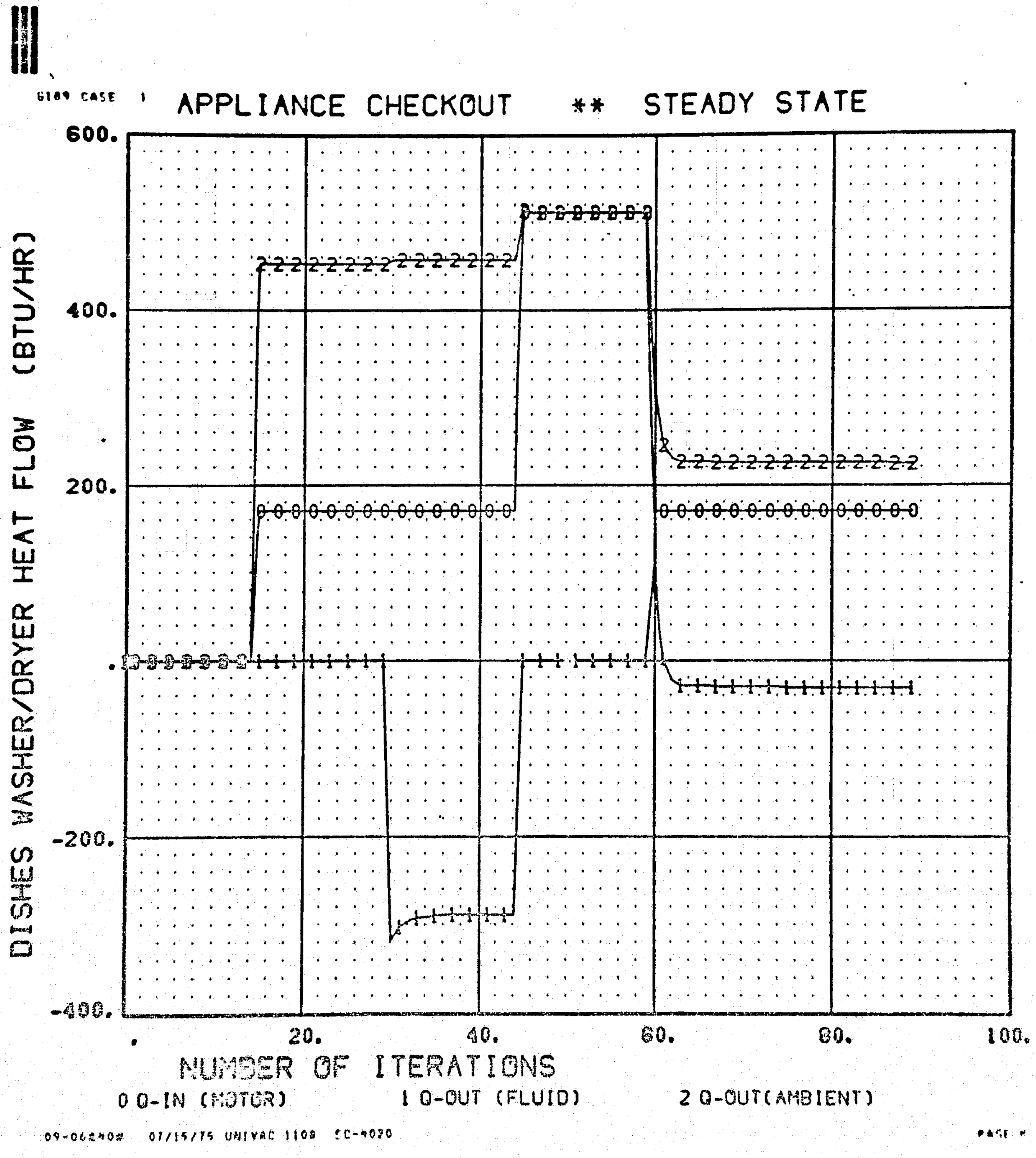

Figure 4-48. Heat Flow for Dishwasher/Dryer Steady-State Checkout Run

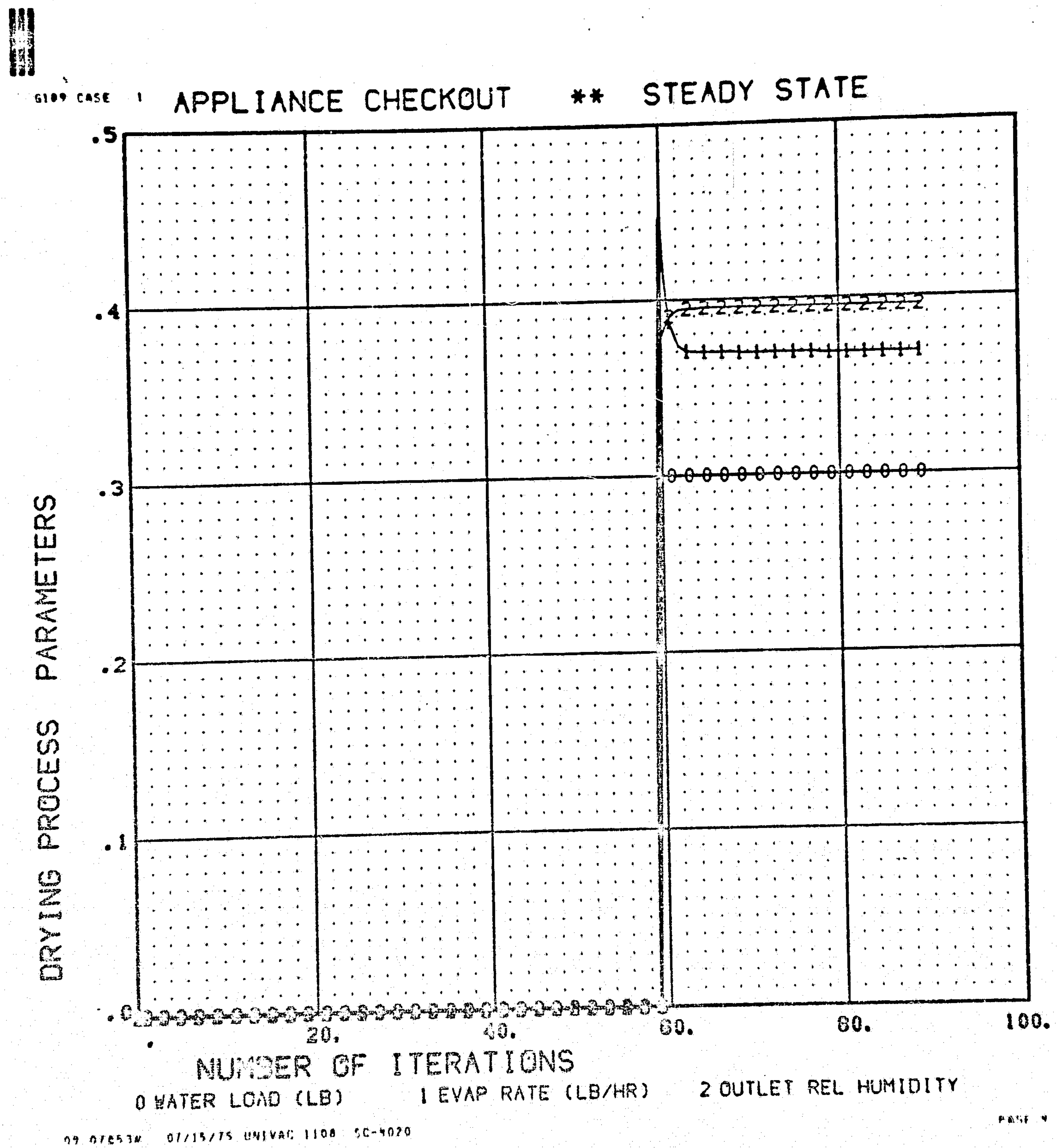

Figure 4-49. Evaporation Parameters for Dishwasher/Dryer Steady-State Checkout Run

TABLE 4-18

FINAL TRANSIENT SOLUTION FOR DISHWASHER/DRYER CHECKOUT RUN

VR(1)=	.00030	VR(2)=	104.44	VR(3)=	.00000	VR(4)=	.00000	VR(20)=	.00000	VR(21)=	104.44
VR(22)=	.00000	VR(23)=	.00000	VR(24)=	.00000	VR(25)=	.00000	VR(26)=	.00000	VR(27)=	.00000
VR(28)=	.00000	VR(29)=	.00000	VR(30)=	.00000	VR(31)=	.00000	VR(32)=	.00000	VR(33)=	.00000
VR(51)=	95.574	VR(52)=	8.0953	VR(53)=	209.09	VR(54)=	70.000	VR(55)=	3.0000	VR(56)=	77.485
VR(57)=	70.000	VR(58)=	1.0000	VR(59)=	28.291	VR(60)=	70.000	VR(61)=	4.0000	VR(62)=	103.31
VR(63)=	95.828	VR(64)=	.00000	VR(65)=	.00000	VR(66)=	15.200	VR(67)=	15.000	VR(68)=	50.000
VR(69)=	.00000	VR(70)=	150.00	VR(71)=	.00000	VR(72)=	104.44	VR(73)=	.00000	VR(74)=	.00000
VR(75)=	.25000	VR(76)=	12.000	VR(77)=	.25000	VR(78)=	.10000-03	VR(79)=	240.00	VR(80)=	10587.
VR(81)=	8.0000	VR(82)=	16.000	VR(83)=	120.00	VR(84)=	.95000	VR(85)=	.00000	VR(86)=	.80000
VR(87)=	.45000	VR(88)=	1.0000	VR(89)=	.40000	VR(90)=	.00000	VR(91)=	.18794	VR(92)=	.00000
VR(93)=	.00000	VR(94)=	.19740-01	VR(95)=	.00000	VR(96)=	.13560-02	VR(97)=	30.000	VR(98)=	30.000
VR(99)=	24.250	VR(100)=	.00000	VR(101)=	.00000	VR(102)=	15.000	VR(103)=	15.000	VR(104)=	.25000-01
VR(105)=	10.000	VR(106)=	4.0000	VR(107)=	10.000	VR(108)=	-1.0000	VR(109)=	.10000-01	VR(110)=	6.0000
VR(111)=	.00000	VR(112)=	.00000	VR(113)=	.00000	VR(114)=	.00000	VR(115)=	.00000	VR(116)=	.00000
VR(117)=	.00000	VR(118)=	.00000	VR(119)=	.00000	VR(120)=	.00000	VR(121)=	.00000	VR(122)=	.00000

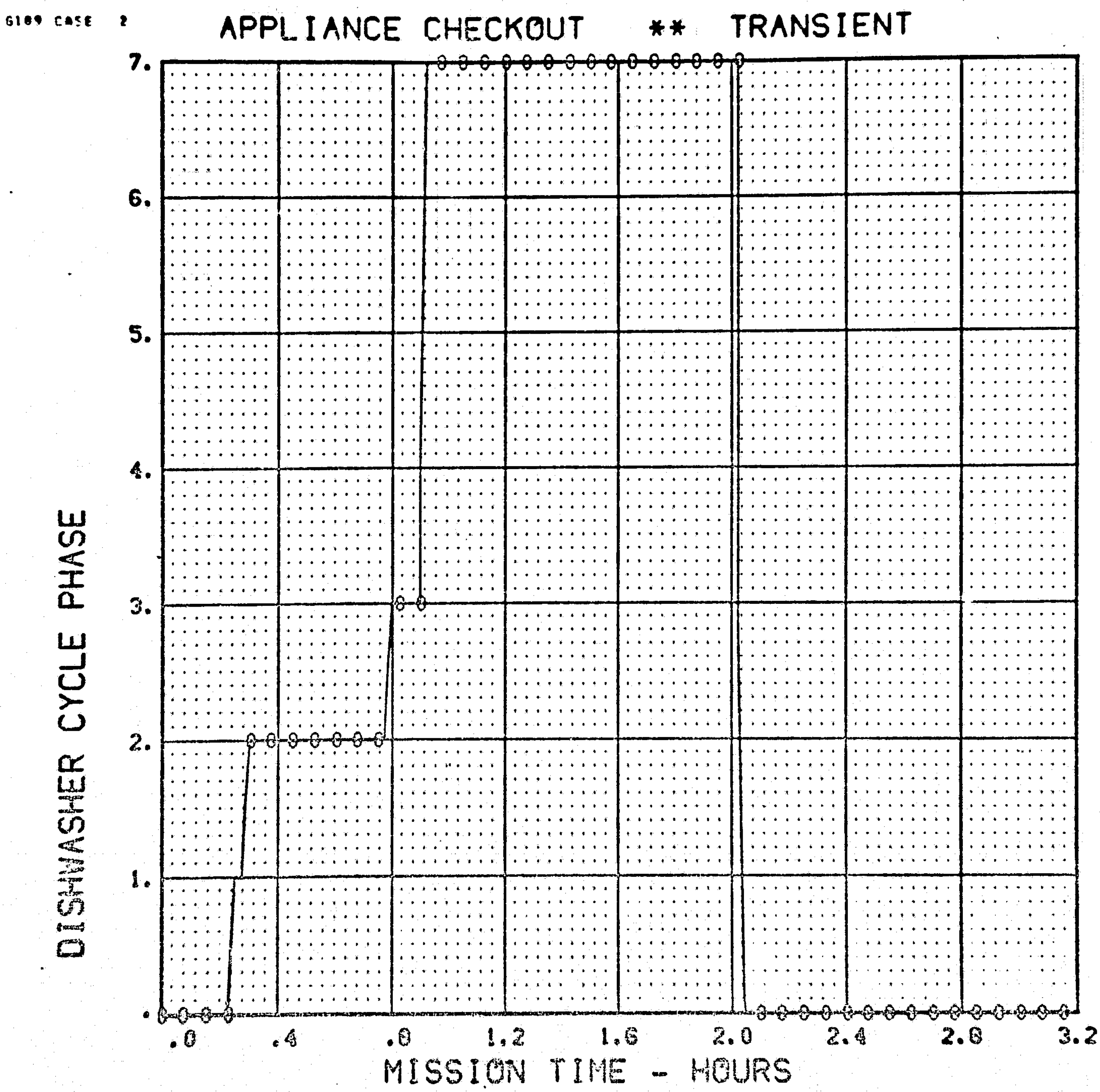

Figure 4-50. Operational Cycle Phase for Dishwasher/Dryer Transient Checkout Run

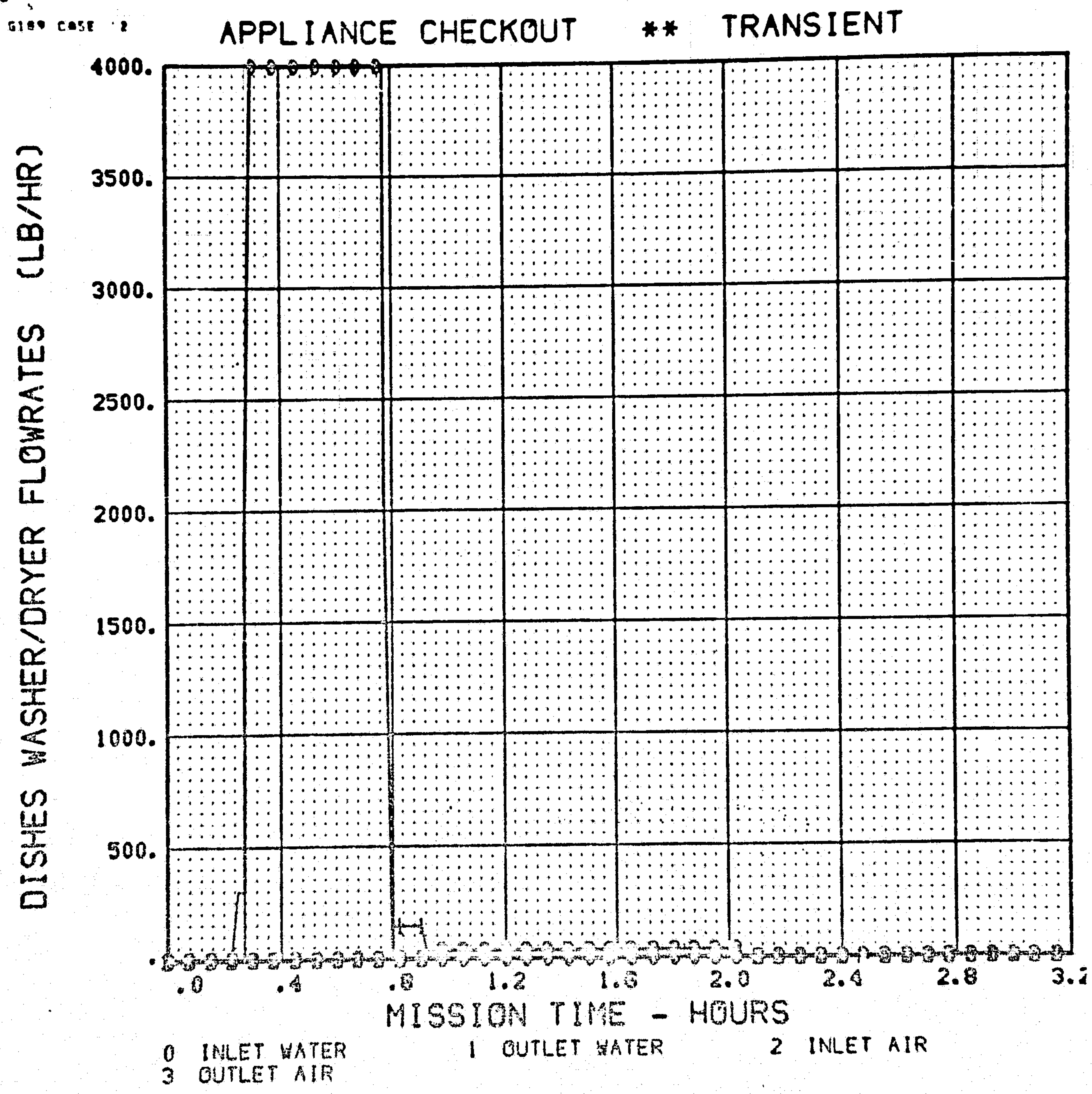

Figure 4-51. Flow Rates for Dishwasher/Dryer Transient Checkout Run

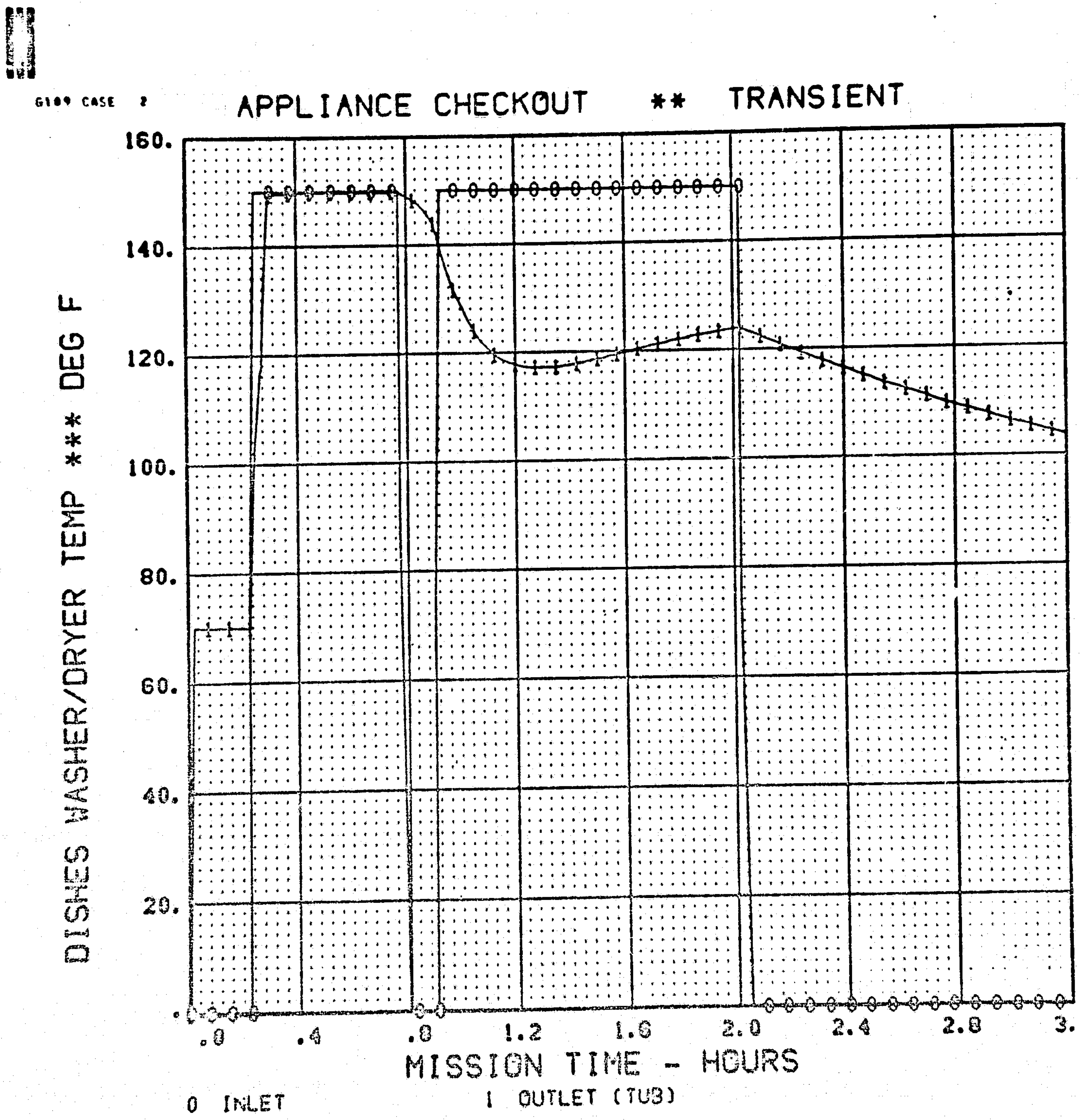

Figure 4-52. Temperatures for Dishwasher/Dryer Transient Checkout Run

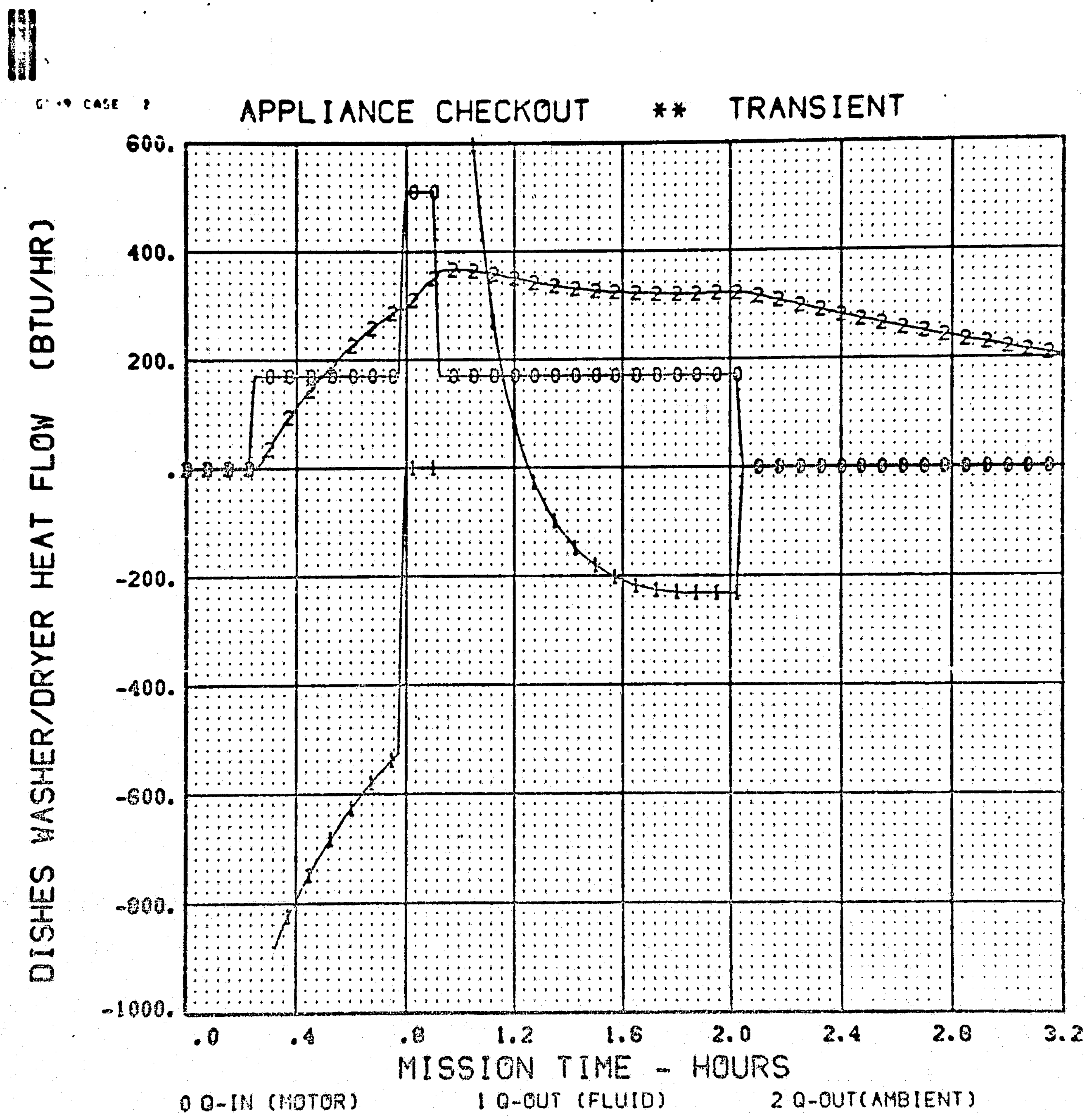

Figure 4-53. Heat Flow for Dishwasher/Dryer
Transient Checkout Run

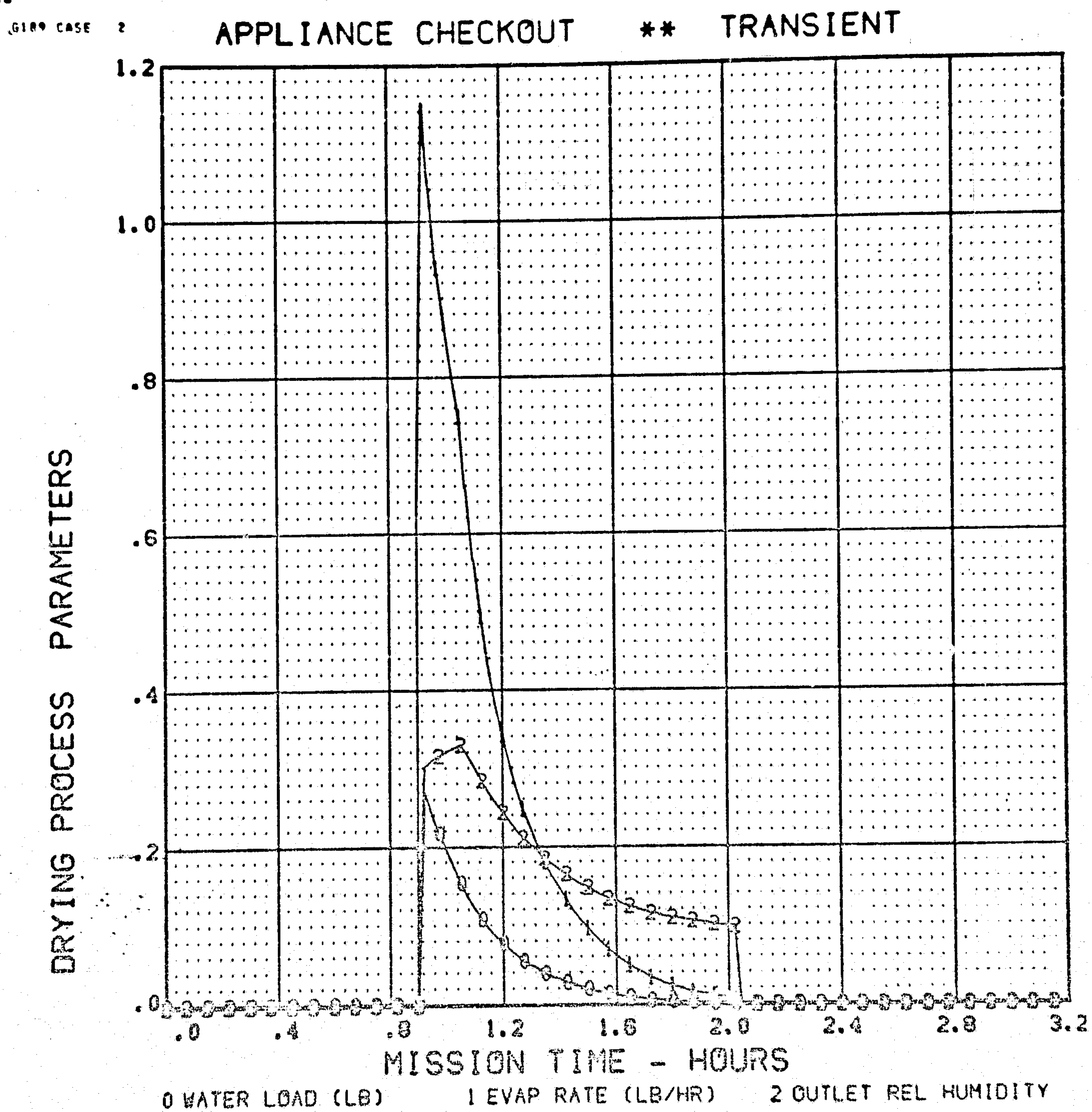

Figure 4-54. Evaporation Parameters for Dishwasher/Dryer Transient Checkout Run

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TABLE 4-19

FINAL STEADY-STATE SOLUTION FOR DRYJOHN CHECKOUT RUN

COMP NO 121 WASTEC SUBR NO 68 PRI SOR 0 SEC SOR 0 COMP PASS NO 99 TIME = .0 SEC

FAILURE FLAGS -- COMP= 0 LOOP= 0

A(1)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPA=	.00000	WTMA=	.00000	RHOA=	.00000	VISCA=	.00000	XKA=	.00000
B(1)=	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000
	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	.00000	
	CPB=	.00000	WTMB=	.00000	RHOB=	.00000	VISCB=	.00000	XKB=	.00000

VR(1)=	.26433-01	VR(2)=	58.134	VR(3)=	.24068	VR(4)=	.24068	VR(5)=	.00000	VR(6)=	.26433-01
VR(7)=	.00000	VR(8)=	.00000	VR(9)=	.10000-29	VR(10)=	.00000	VR(11)=	.00000	VR(12)=	.00000
VR(13)=	.00000	VR(14)=	.00000	VR(20)=	.00000	VR(21)=	.00000	VR(22)=	.00000	VR(23)=	.00000
VR(24)=	.00000	VR(25)=	.00000	VR(26)=	.00000	VR(27)=	.00000	VR(28)=	.00000	VR(29)=	.00000
VR(30)=	.00000	VR(31)=	.00000	VR(32)=	.00000	VR(33)=	.00000	VR(51)=	58.134	VR(52)=	2.3212
VR(53)=	-27.544	VR(54)=	70.000	VR(55)=	2.0000	VR(56)=	-11.191	VR(57)=	70.000	VR(58)=	.80000
VR(59)=	-4.4868	VR(60)=	70.000	VR(61)=	1.0000	VR(62)=	-11.866	VR(63)=	64.405	VR(64)=	2.5000
VR(65)=	.00000	VR(66)=	.00000	VR(67)=	20.000	VR(68)=	.00000	VR(69)=	108.00	VR(70)=	.00000
VR(71)=	.00000	VR(72)=	.00000	VR(73)=	.00000	VR(74)=	1.0000	VR(75)=	2.5000	VR(76)=	.26433-01
VR(77)=	1.5000	VR(78)=	.15000	VR(79)=	97.000	VR(80)=	3.0000	VR(81)=	1.2000	VR(82)=	.00000
VR(83)=	.60000	VR(84)=	.00000	VR(85)=	.50000	VR(86)=	.30000	VR(87)=	.75000	VR(88)=	.25000
VR(89)=	.20000	VR(90)=	.00000	VR(91)=	2.5000	VR(92)=	.00000	VR(93)=	.00000	VR(94)=	.00000
VR(95)=	.00000	VR(96)=	.00000	VR(97)=	.00000	VR(98)=	.00000	VR(99)=	.00000	VR(100)=	.00000
VR(101)=	1.00000	VR(102)=	.00000	VR(103)=	.00000	VR(104)=	.80000-01	VR(105)=	.00000	VR(106)=	.00000
VR(107)=	.00000	VR(									
		CPP=	.44000	WTMP=	18.000	RHOP=	.77929-03	VISCP=	.47000-01	XKP=	.15400-01
		CPS=	.00000	WTMS=	.00000	RHOS=	.00000	VISCS=	.00000	XKS=	.00000

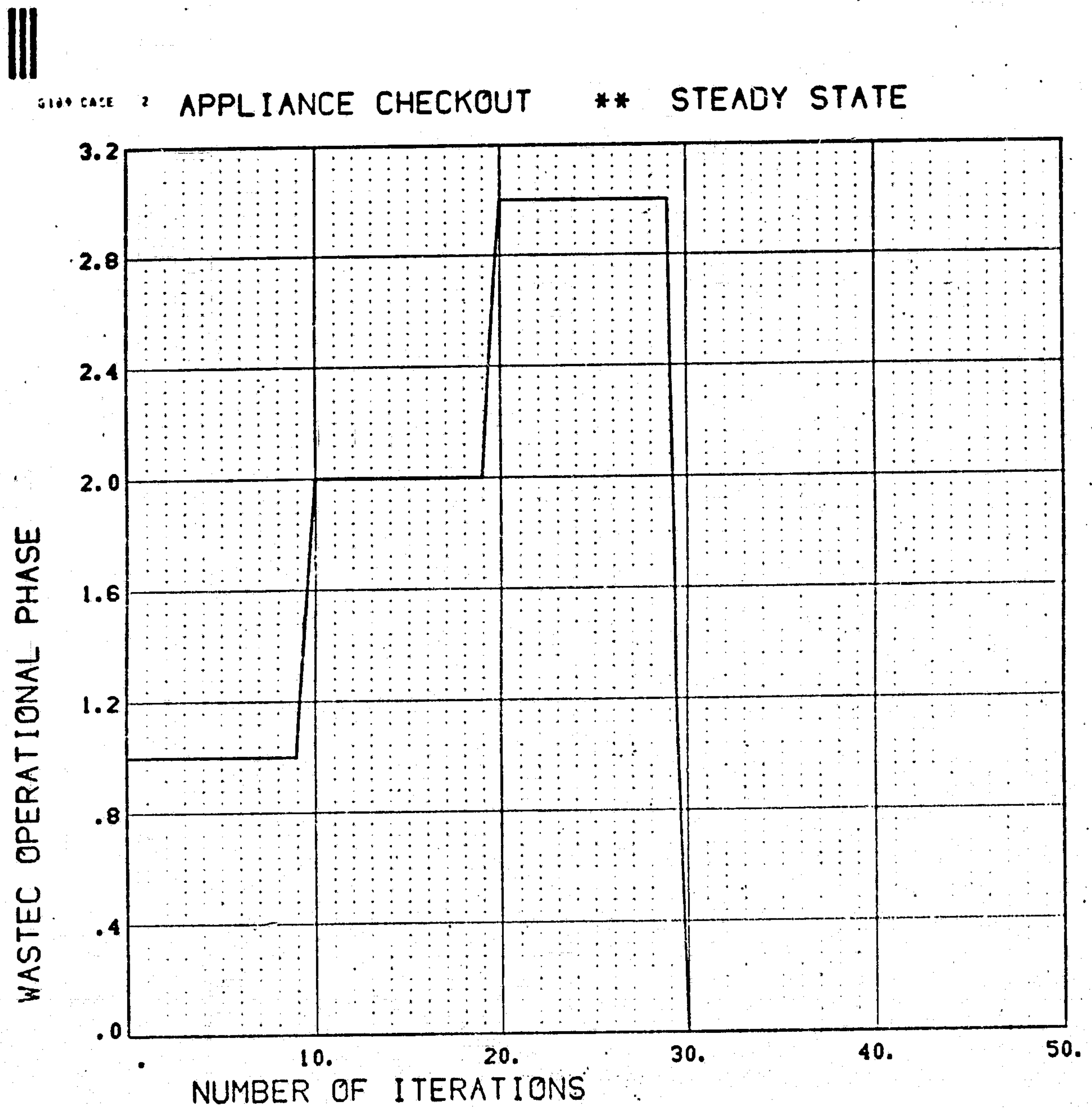

Figure 4-55. Usage Phase for Dryjohn Steady-State Checkout Run

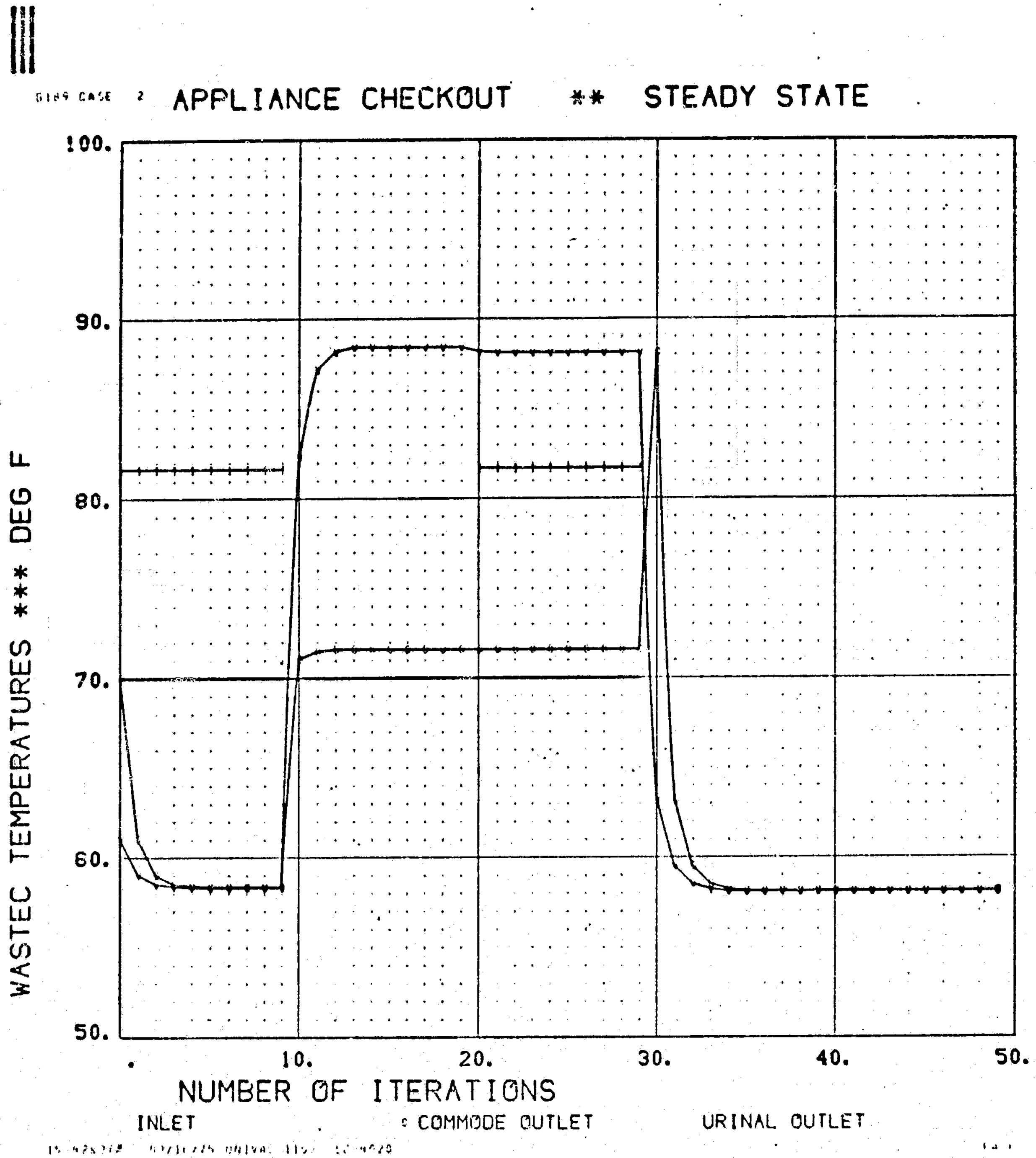

Figure 4-56. Temperatures for Dryjohn Steady-State Checkout Run

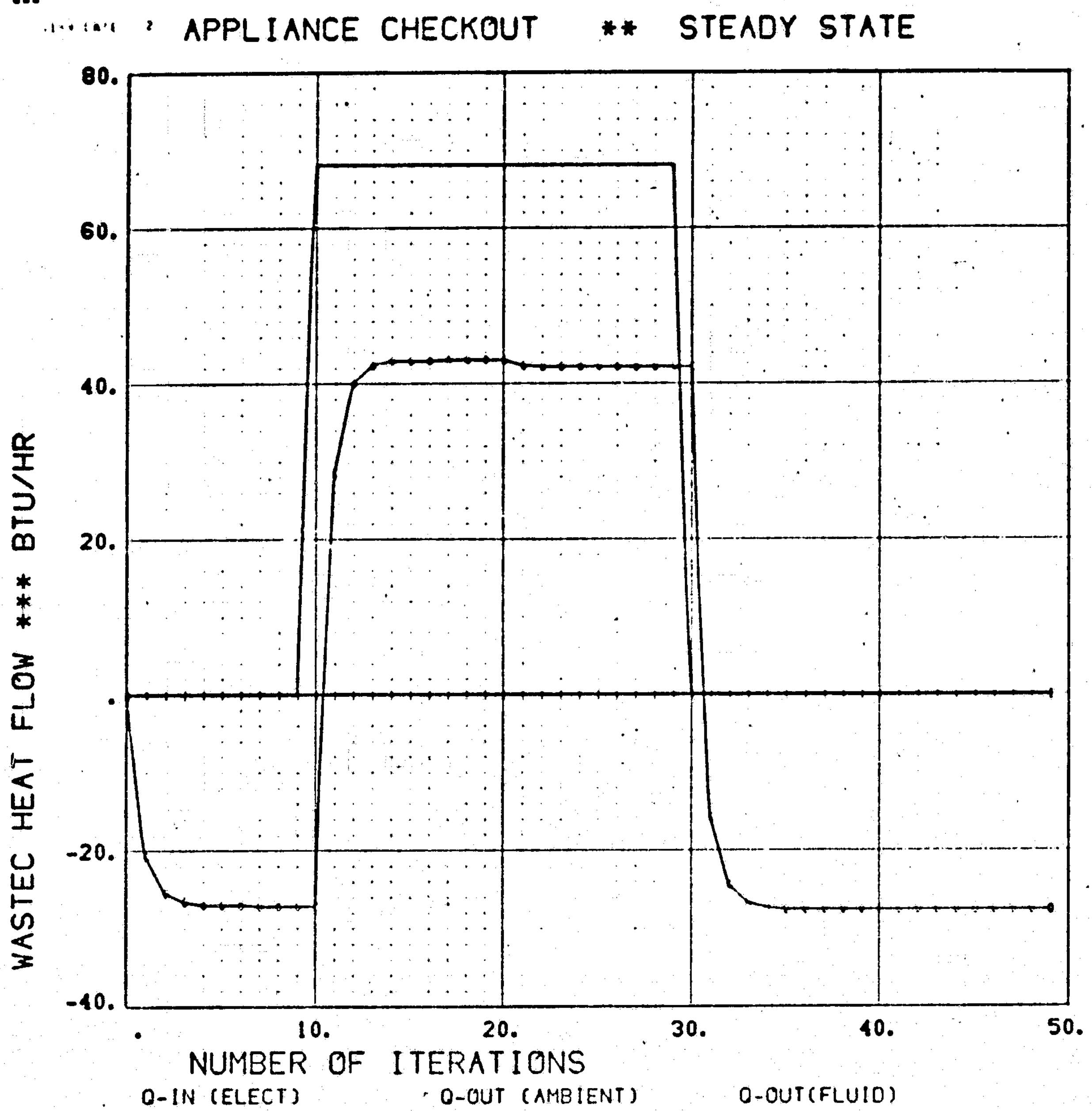

Figure 4-57. Heat Flow for Dryjohn Steady-State Checkout Run

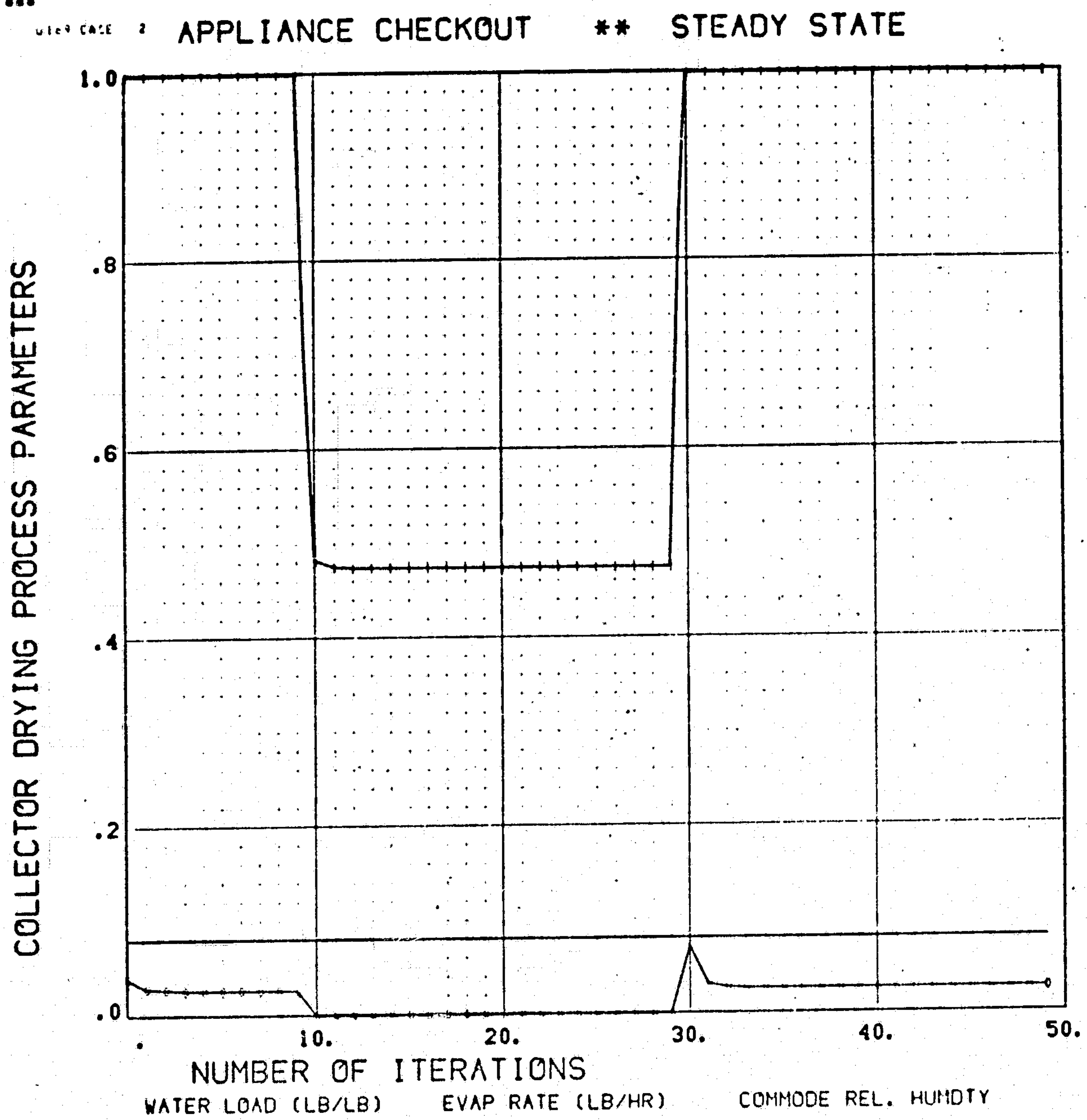

Figure 4-58. Collector Evaporation Parameters for Dryjohn Steady-State Checkout Run

4.6.1 (Continued)

PHASE 1 - Urine Collection

The first steady-state case run was phase 1 (urine collection). The subroutine assumes a urinal flow addition equal to the micturition flow rate which was input in R(69) as 108 lb/hr. The default value of 97°F was used for the urine inlet temperature. The inlet air flow rate was 85.2 lb/hr total at 70°F, of which 0.665 lb/hr was water vapor. The outlet water vapor flow rate was 2.007 lb/hr. Therefore, 1.342 lb/hr of water were evaporated in the urine/air mixing process. The latent cooling from this source would be approximately 1.342 x 1060 = 1422.5 Btu/hr. The heat conducted from the fluid to the dryjohn (through conductor G_u in Figure 3-16) is given by

$$q_{\text{urinal fluid to dryjohn}} = G_u \left(T_{\text{urinal out}} - T_{\text{dryjohn}}\right)$$

$$= R(78)\ [R(21) - R(51)]$$

$$= 0.15\ \text{Btu/hr}\ {}^\circ\text{F}\ (81.7^\circ\text{F} - 58.4^\circ\text{F})$$

$$= 3.5\ \text{Btu/hr}$$

Therefore, the net heat out of the urinal fluid mixture was 1422.5 + 3.17 = 1426 Btu/hr. This should correspond to the predicted sensible heat loss and corresponding temperature drop of the air and urine. This is given as follows:

$$q_{\text{air sensible loss}} = \dot{m}\ c_p\ \Delta T = (85.2\ \text{lb/hr})(0.243\ \text{Btu/lb}^\circ\text{F})(70^\circ\text{F}-81.7^\circ\text{F})$$

$$= -242.2\ \text{Btu/hr}$$

$$q_{\text{urine sensible loss}} = \dot{m}\ c_p\ \Delta T = (108\ \text{lb/hr})(1\ \text{Btu/lb}^\circ\text{F})(97 - 81.7)$$

$$= 1652.4\ \text{Btu/hr}$$

4.6.1 (Continued)

Thus, the net sensible heat loss of the fluid is about 1652.4 - 242.2 = 1410 Btu/hr. This is approximately equal to the required heat loss shown previously to be 1426 Btu/hr. The reason for the difference is that a more accurate and slightly different method is used in the subroutine than the simplified equations given previously to check the results. However, the agreement is within the limits of normal uncertainties in the model input data, and the urinal heat balance and outlet temperature are valid.

The collector contents were assumed to be under vacuum during this phase. Their temperature is given in Figure 4-56, and it is only slightly warmer (0.2°F) than the solution shown for phase 0. The results for that phase are discussed later. The 0.2°F temperature increase during phase 1 is caused by the heat transferred in from the urinal which is in use.

PHASE 2 - Fecal Collection

During this phase, 68.2 Btu/hr are input to the collector from the 20 watt slinger motor as reflected in Figure 4-57. In steady state, all this heat should be dissipated to the air and structure. The heat input to the 63.9 lb/hr collector air flow is given by

$$q_{air} = \dot{m}\, c_p\, \Delta T = (63.9\ \text{lb/hr})(0.243\ \text{Btu/lb°F})(71.63°\text{F}-70°\text{F})$$

$$= 25.0\ \text{Btu/hr}$$

The heat loss through the dryjohn structure to ambient is given by

$$q_{structure} = G\, \Delta T = (2.33\ \text{Btu/hr °F})(88.47°\text{F} - 70°\text{F})$$

$$= 43.0\ \text{Btu/hr}$$

4.6.1 (Continued)

Thus, the total heat loss to the air and structure is 25 + 43 = 68 Btu/hr which is equal, as shown previously, to the total heat input as required. No evaporative cooling is involved during this phase since the collector is not under vacuum.

PHASE 3 - Combined Urine/Fecal Collection

The urinal temperatures for this case are shown in Figure 4-56 to be approximately equal to those described previously for phase 1 (urine collection) as required. The collector temperatures are shown in the same figure to be equal approximately to those described previously for phase 2 (fecal collection) as expected. The heat flow in Figure 4-57 is also approximately equal to that for phase 2 as expected.

PHASE 0 - Unit Off

The results for phase 0 (unit off) with collector being vacuum dried were included at the end of the run as shown in Figures 4-55 through 4-58. The collector contents are shown in Figure 4-56 to be at 58.1°F. As seen in Figure 4-58, the collector outlet is saturated; that is, only water vapor is being outgassed from the fecal material. Water vapor pressure at 58.1°F is approximately 0.24 psia. The evaporation rate, from equation (3.6.6), should therefore be

$$\dot{m}_{evap} = \frac{f\ A_c\ P_\omega}{\sqrt{T_c}}$$

$$= \frac{(2.5)(1.0\text{ sq ft})(0.24\text{ psia})}{\sqrt{460 + 58.1°F}}$$

$$= 0.0264\text{ lb/hr}$$

4.6.1 (Continued)

As seen in Figure 4-58 and Table 4-19, this rate is evaluated correctly. From actual dryjohn data, Reference 28, it is reported that 4 hours are required for new fecal material to lose one-half of their initial water content. Assuming 0.3 lb fecal mass per defecation with 75 percent water content, this would require an average evaporation rate of 0.028 lb/hr to lose one-half its moisture in 4 hours. Of course, this rate will vary widely with time and with different fecal properties, and the rate predicted in the subroutine is within a typical range of operating conditions. For the evaporation rate of 0.0264 lb/hr, the cooling effect on the dryjohn should be (0.0264 lb/hr) x (1042 Btu/lb) = 27.5 Btu/hr. This value is correctly reflected in Figure 4-58. This should also be equal to the heat conducted into the collector structure, as given by

$$q_{structure} = G_{effective} \Delta T = (2.32\ \text{Btu/hr}^\circ\text{F})(70^\circ\text{F}-58.13^\circ\text{F})$$

$$= 27.5\ \text{Btu/hr}$$

Thus, the energy into and out of the collector is equal, as required, and the computed collector contents temperature is valid.

4.6.2 <u>Dryjohn Transient Run</u>

The input model data for this case were identical to the data used for the steady-state case. The final solution is tabulated in Table 4-20. Four usage phases were run: (1) urine collection, (2) fecal collection, (3) combined urine/fecal collection, and (0) unit off with vacuum dry. The transient results are plotted in Figures 4-59 through 4-62. These results may be compared with the steady-state solution described previously. The primary difference between the two solutions is the urinal outlet temperature during phases 1 and 3. For steady state, a continual

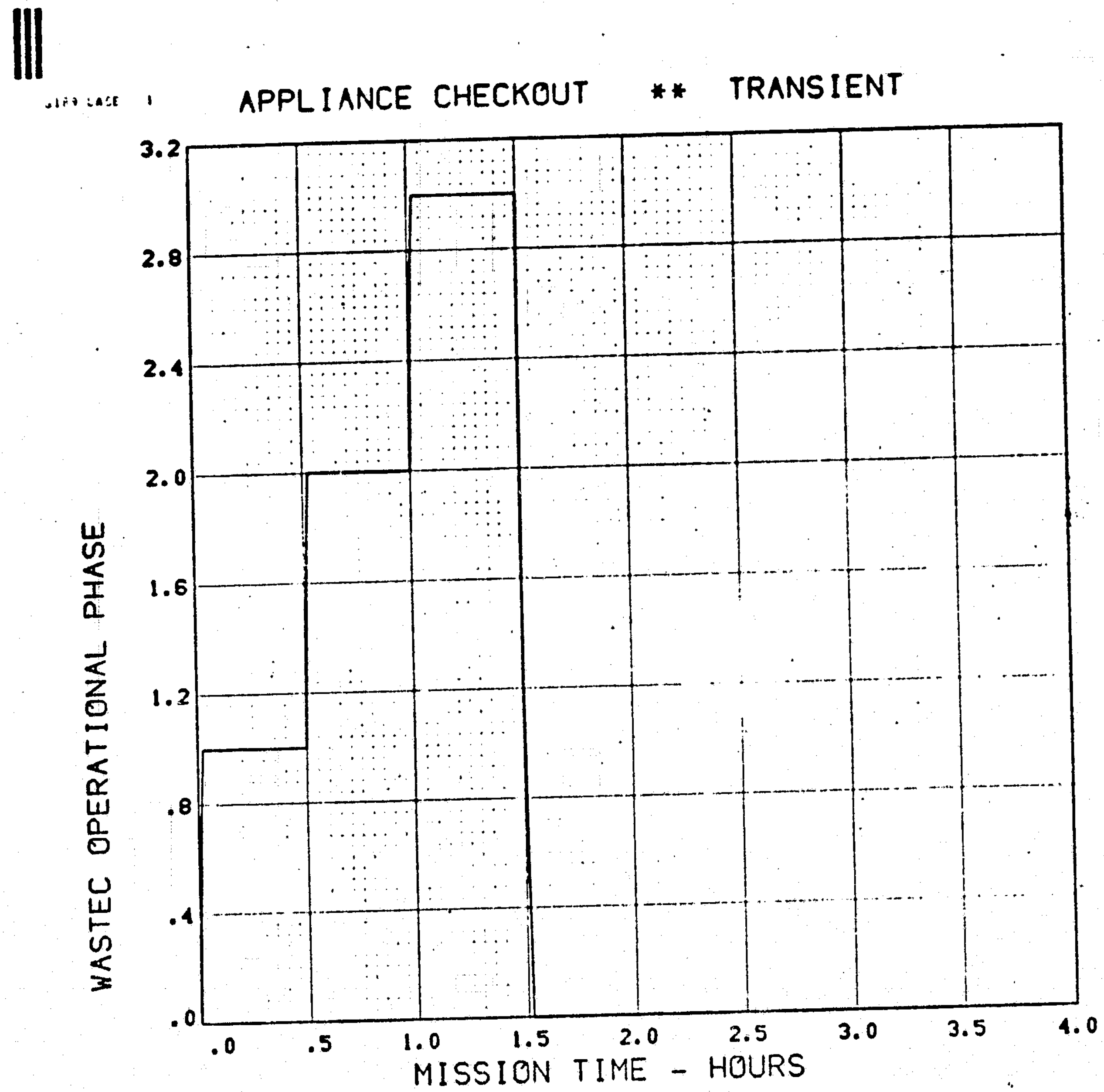

Figure 4-59. Usage Phase for Dryjohn Transient Checkout Run

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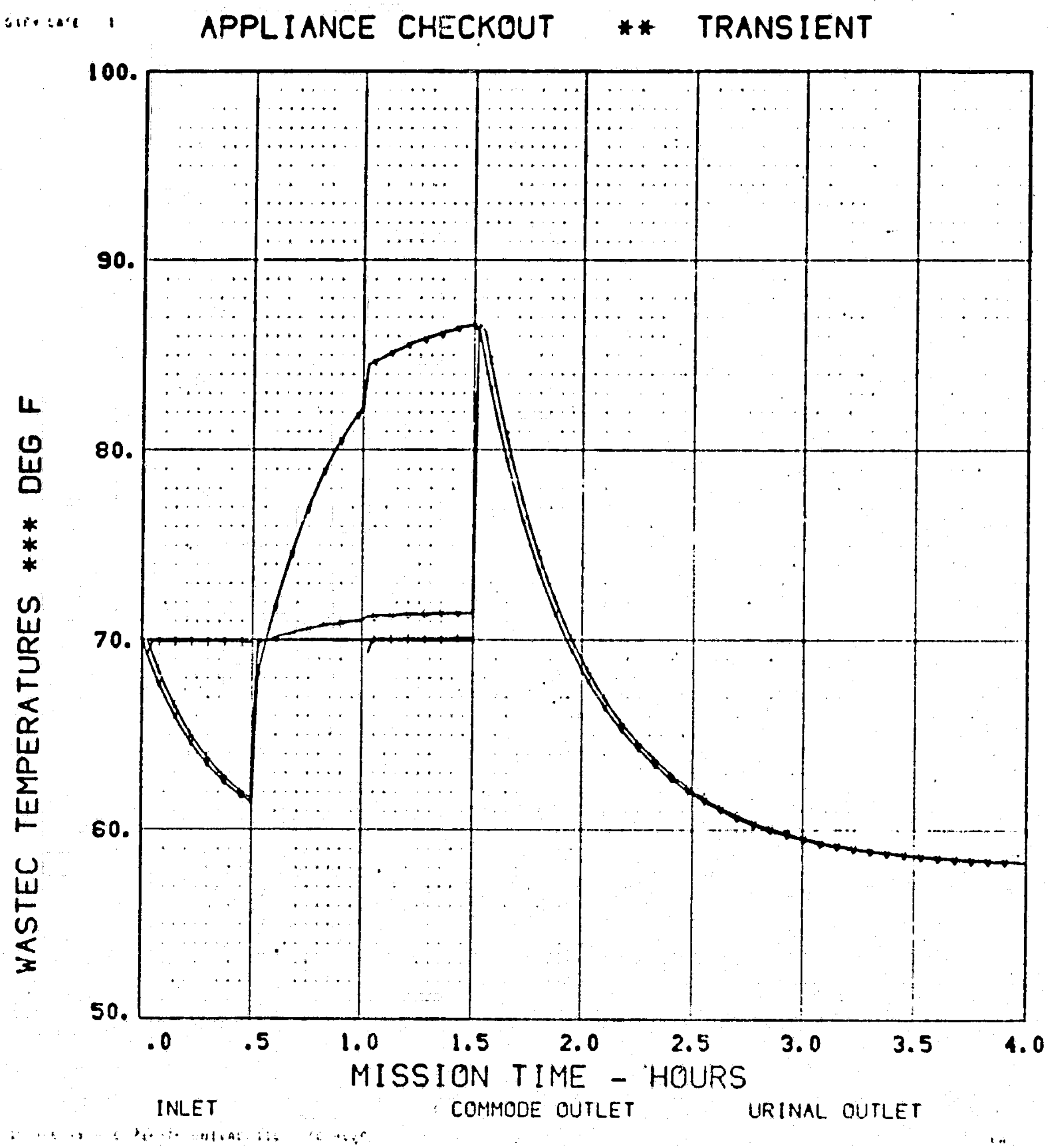

Figure 4-60. Temperatures for Dryjohn Transient Checkout Run

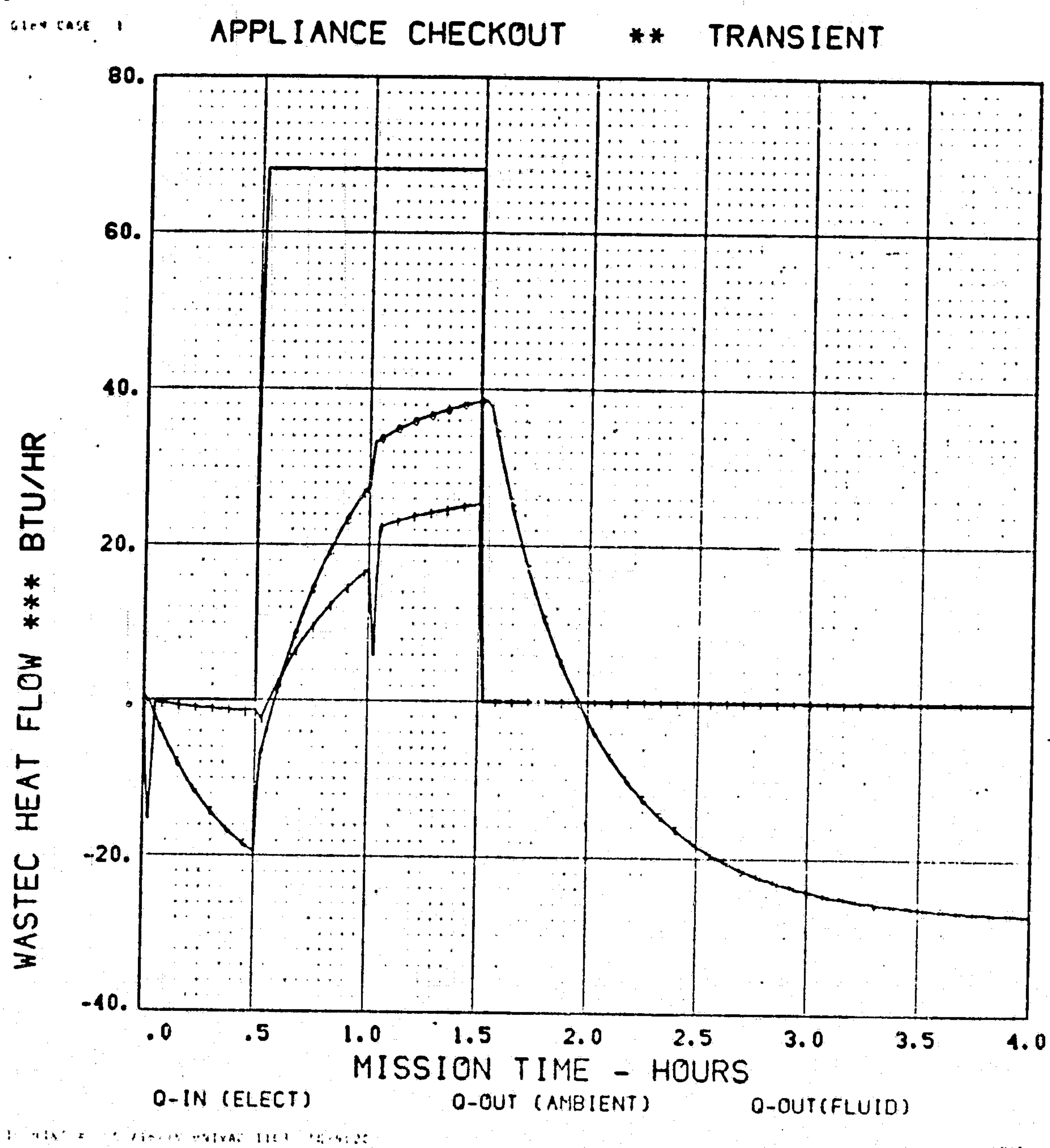

Figure 4-61. Heat Flow for Dryjohn Transient Checkout Run

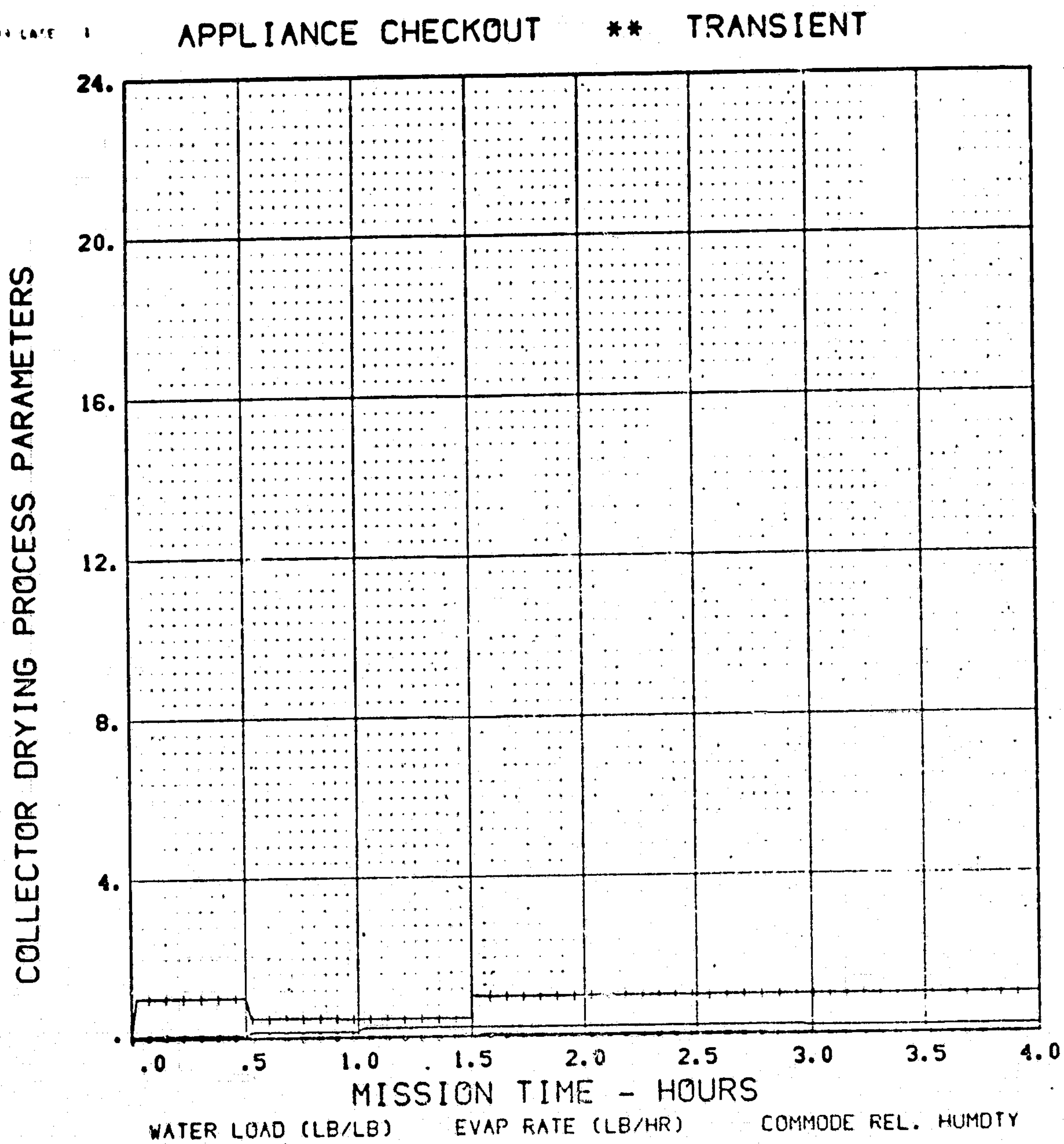

Figure 4-62. Collector Evaporation Parameters for Dryjohn Transient Checkout Run

4.6.2 (Continued)

urine outlet flow equal to the input micturition flow rate was assumed. For the transient run, however, this flow was limited to the average urine generation of 0.6 lb/use which was input, and then it was stopped. Consequently, the urinal outlet temperature did not rise appreciably above ambient.

5.0 SHUTTLE ORBITER APPLIANCES SIMULATION

To demonstrate that the appliance subroutines are operational in an all-up G-189A system simulation, they were run in a Space Shuttle Orbiter and Modular Space Station G-189A ECLSS model. The appliances included in the models were taken from trade studies of many candidate concepts (Reference 1) and represent the optimum set of appliances for each vehicle. The results of these trade studies for a four-man, 20.5-day Shuttle Orbiter are shown in Table 2-1. Due to the short mission duration, the most simple concepts were selected in most cases. The only Shuttle appliances selected which required the new G-189A appliance subroutines were a space radiator refrigerator, food heating/serving trays, and a dryjohn. Since a refrigerator is currently not included in the Shuttle design and since the Shuttle cabin coolant loop would be only marginally effective for the space radiator concept, the refrigerator was not included in the Shuttle model. Both refrigerators and freezers are included in the Space Station model in Section 6. A food and medical sample freezer has been developed for use on Shuttle, as described in Reference 32. This freezer is not included in the Shuttle model. However, the CHILLR subroutine has been adapted to simulate the freezer, as described in Sections 3.1.1 and 4.1.5.

5.1 UNMODIFIED SHUTTLE CASE

An available steady-state G-189A model of the Shuttle Orbiter ECLSS (described in Reference 3), together with several model modifications supplied by Mc Donnell Douglas Corporation (MDAC), was used for the basic Shuttle system model. This model obtains a single steady-state solution for up to 23 different mission phases. For the purpose of verifying the appliance subroutines, this model was run for mission phases 12, 13, 14, and 15 which represent four different days of normal orbital operations. Two cases were run. First, an unmodified case was run with all program inputs exactly as supplied by MDAC. No appliances or modifications were included. The input model data and GPOLY subroutines for this run are included in Appendix B. The output tables and plots from this run are listed in Appendix C.

5.2 SHUTTLE CASE WITH APPLIANCES ADDED

A dryjohn and food heating trays were then added to the basic Shuttle model described previously, and the same case was rerun. The changes made in the basic model and GPOLY subroutines are listed in Appendix D. A flow schematic of the G-189A components added to the basic Shuttle case is shown in Figure 5-1. The dryjohn air inlet and outlet were connected directly to the secondary side of component 2, which was the Orbiter cabin in the basic model. The dryjohn collector and urinal outlets (component 121) were directed to a water separator (component 125). This water separator also had a fluid inlet from the airlock to simulate the actual Shuttle configuration. However, since the basic Shuttle model used did not have the airlock included, a dummy input (component 130) was used for this connection with a flow rate of zero to simulate normal crew work activities. Another dummy input from the air revitalization system waste water (component 130) to the waste water storage tank (component 129) was also included to make the model complete, although it was also given a flow rate of zero. The heat conducted through the dryjohn structure to the cabin (stored in the dryjohn R-array location 53) was added in GPOLY1 to the total cabin input heat load in cabin component R-array location 66. The waste water storage tank (component 129) was included in the model. However, the heat dissipated through the tank structure was not added to the cabin heat load since the tank model, not being part of the dryjohn system, was not sufficiently correlated to accurately simulate the actual Shuttle tanks.

The food heating trays were not connected directly in any fluid flow loop; thus, no components are shown attached in Figure 5-1. The heat dissipated from the food trays to the cabin (stored in the food tray R-array location 53) was added in GPOLY1 to the total cabin heat load in cabin component R-array location 66.

To exercise various appliance subroutine options, different conditions were assumed during each of the four mission phases. These are shown in Table 5-1. The output tables and plots from this run are listed in Appendix E. The

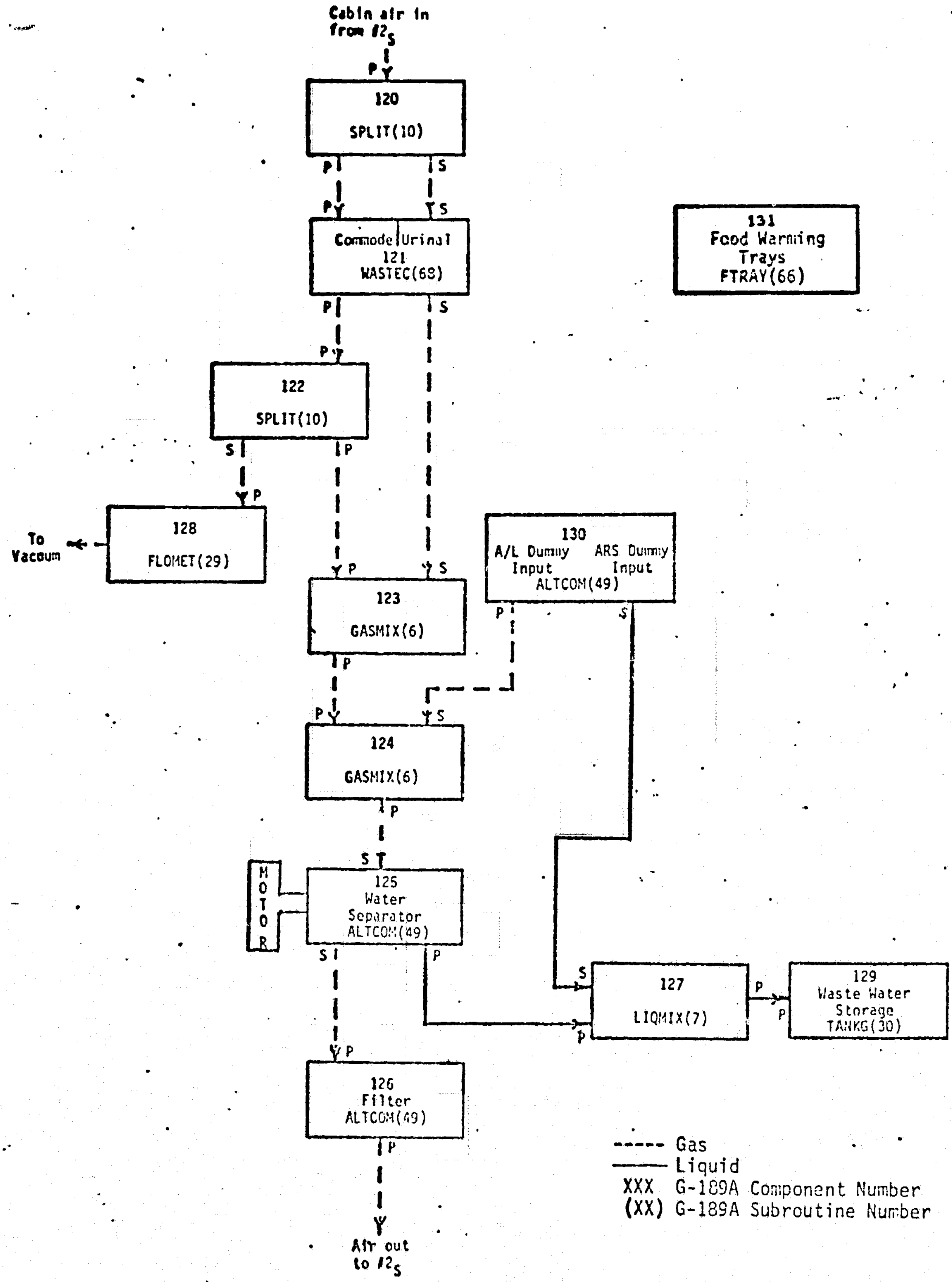

Figure 5-1. Flow Schematic of G-189A Appliance Components Added to Basic Shuttle Orbiter Model

5.2 (Continued)

identical parameters were plotted here which were plotted previously in Appendix C for the unmodified case. All differences between the two solutions are attributable to the addition of the dryjohn and food trays. In addition, plots were made of the dryjohn and food trays performance during the four mission phases in Figures 5-2 to 5-10. In all cases, they are seen to reach approximately the same solutions which were presented for the individual subroutines in Paragraphs 4.2 and 4.6, thus demonstrating that the new subroutines are working properly in an all-up G-189A system model.

TABLE 5-1

APPLIANCE COMPONENT OPTIONS ASSUMED
IN SHUTTLE ORBITER SIMULATION

	D R Y J O H N				FOOD TRAYS
MISSION	USAGE		CABIN AIR FLOW,CFM		TOTAL NUMBER
PHASE	PHASE	DESCRIPTION	URINAL	COLLECTOR	BEING HEATED
12	0	Not in use Vacuum dry	0	0	0
13	1	Urine collection Vacuum dry	20	0	2
14	2	Fecal collection	20	15	3
15	3	Combined urine/ fecal collection	20	15	4

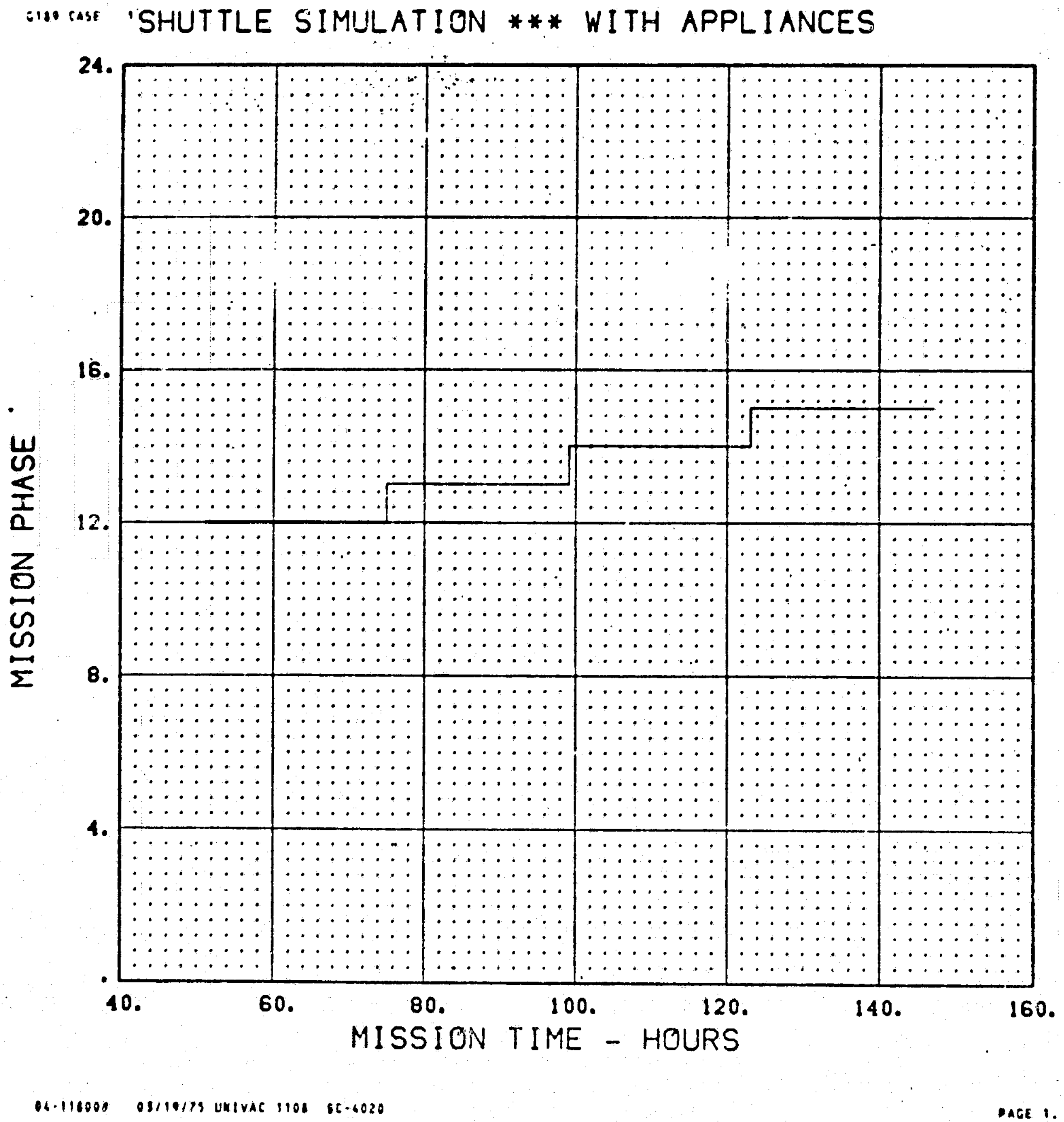

Figure 5-2. Mission Phase for Shuttle Appliances Simulation

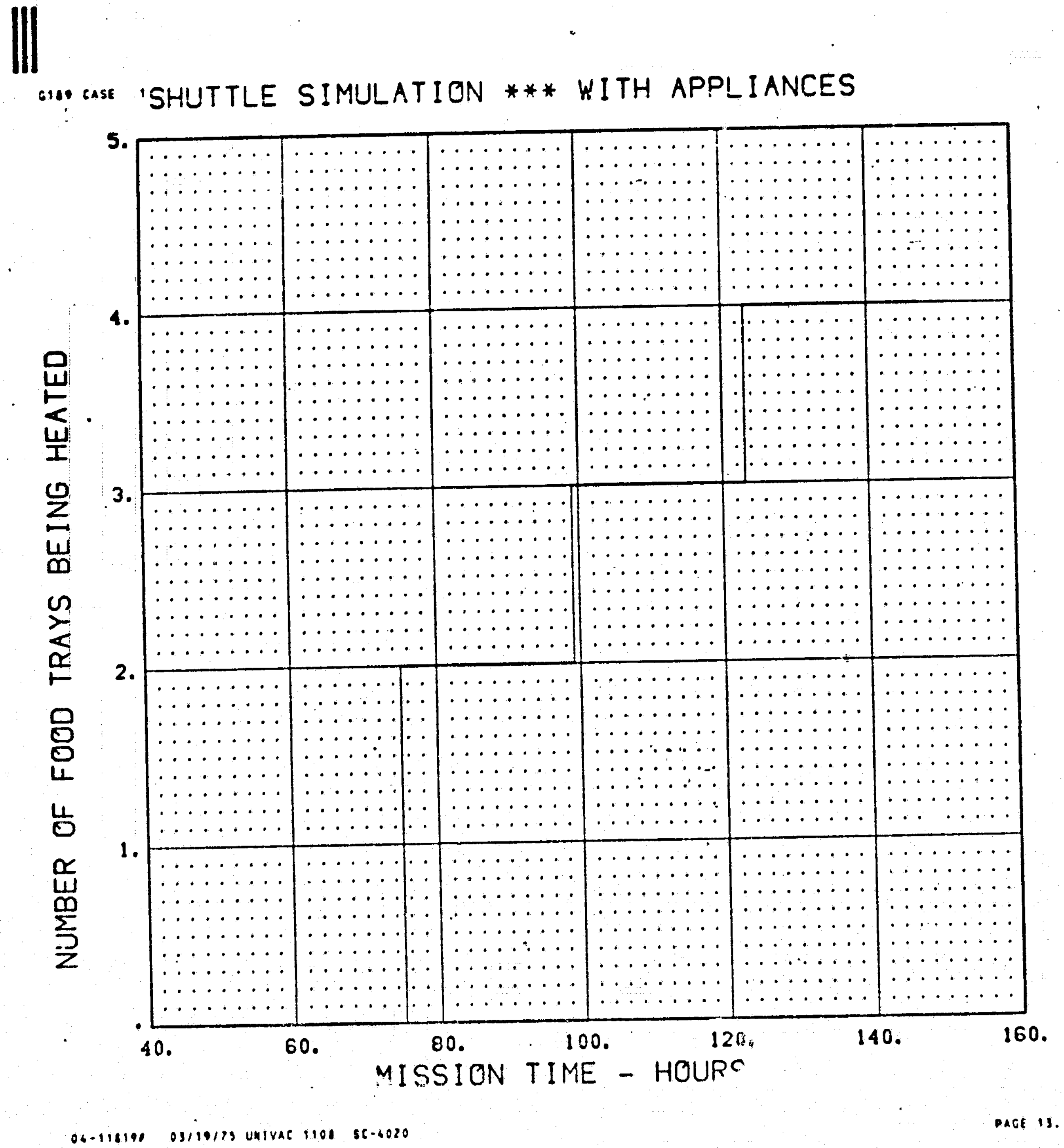

Figure 5-3. Number of Heated Food Trays in Shuttle Appliances Simulation

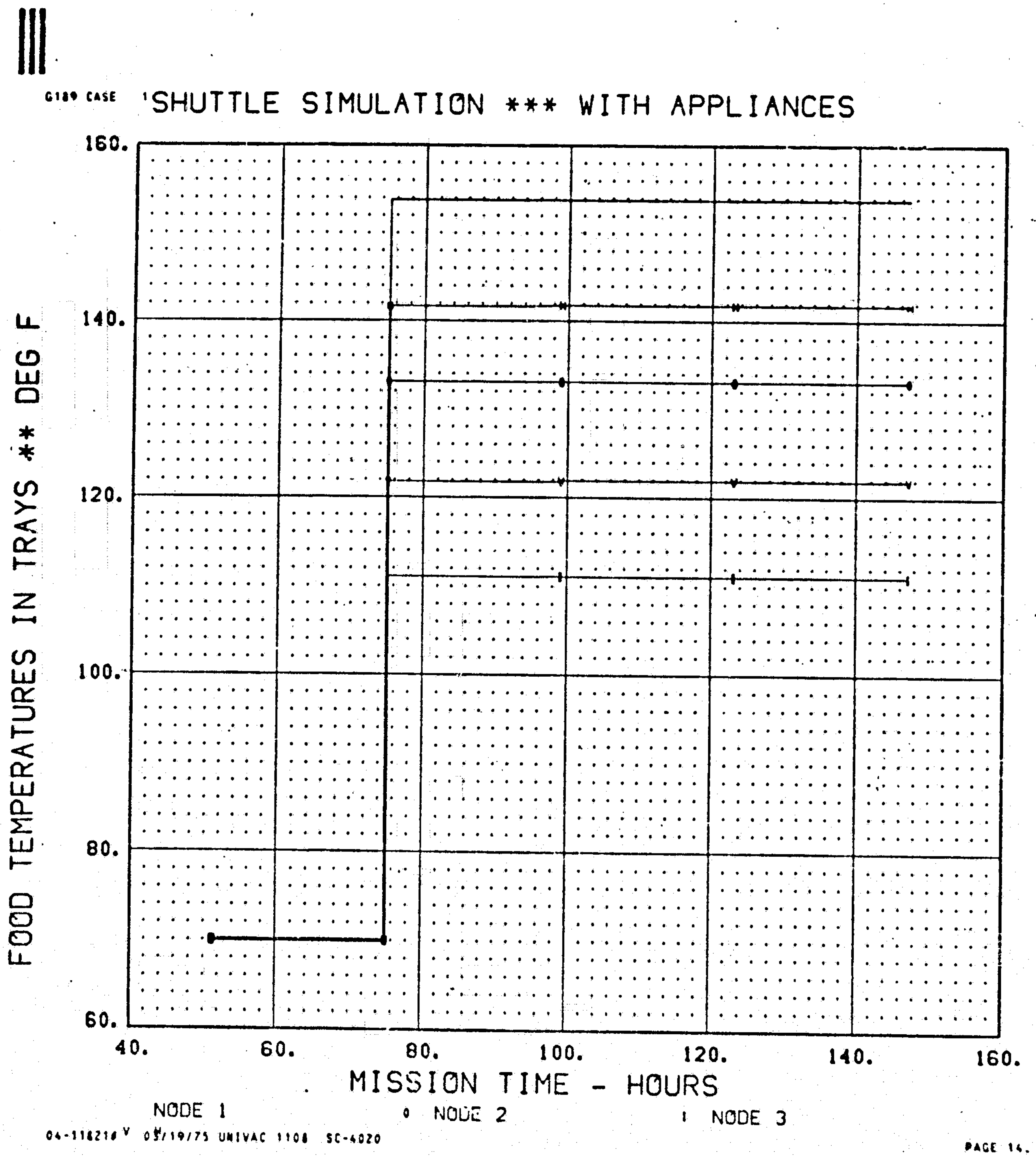

Figure 5-4. Food Tray Temperatures in Shuttle Appliances Simulation

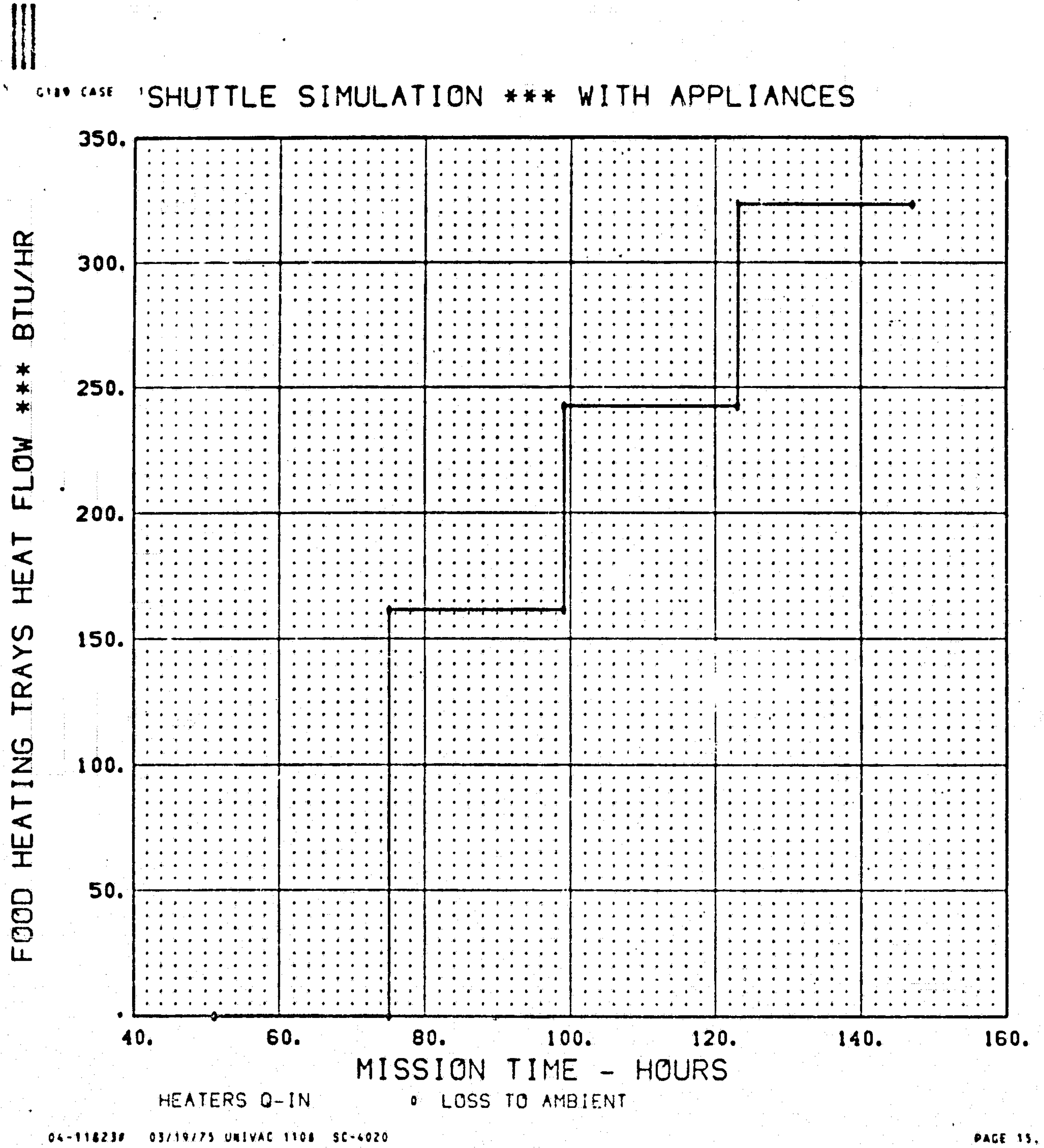

Figure 5-5. Food Trays Heat Loss in Shuttle Appliances Simulation

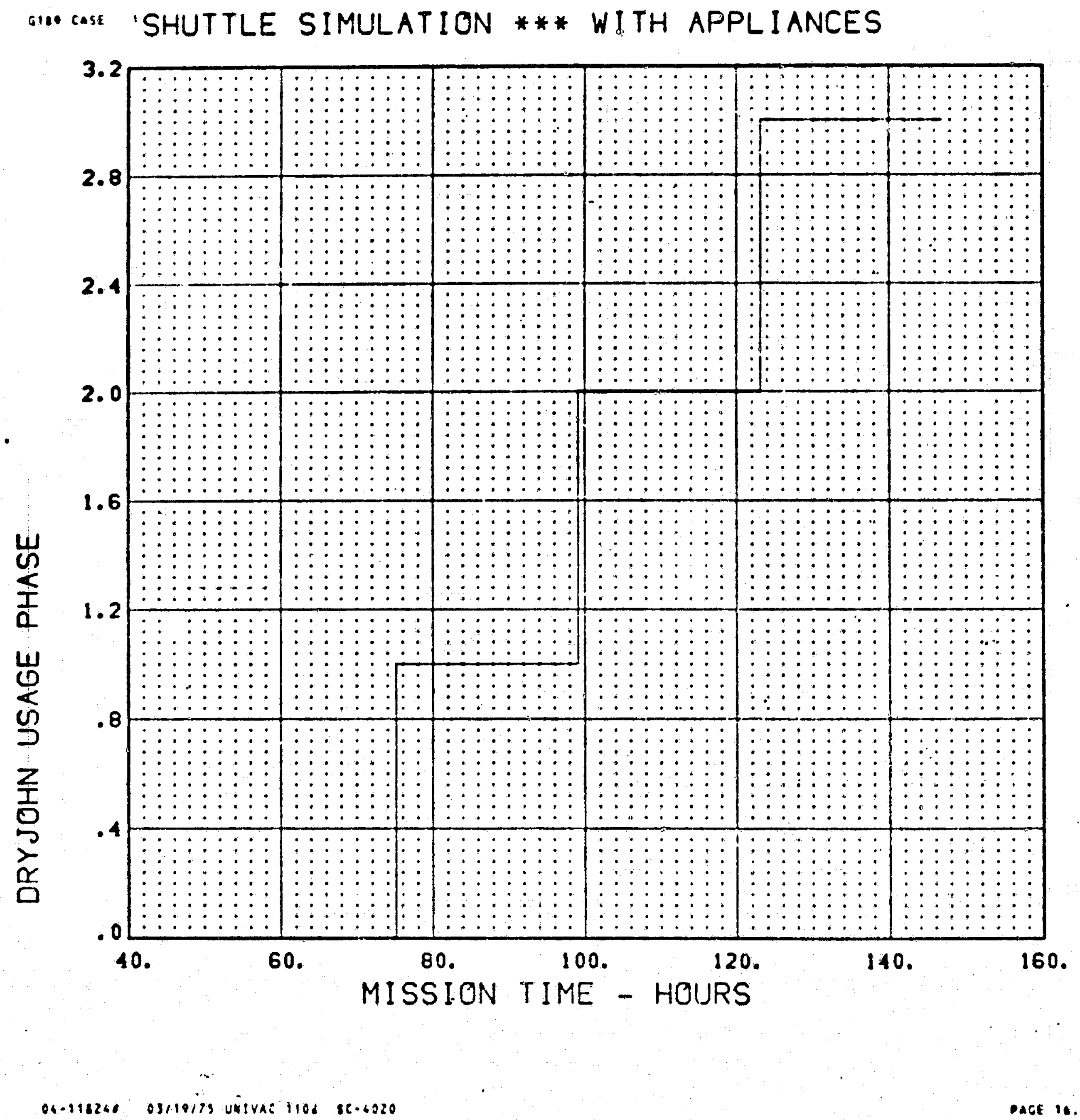

Figure 5-6. Dryjohn Usage Phase in Shuttle Appliances Simulation

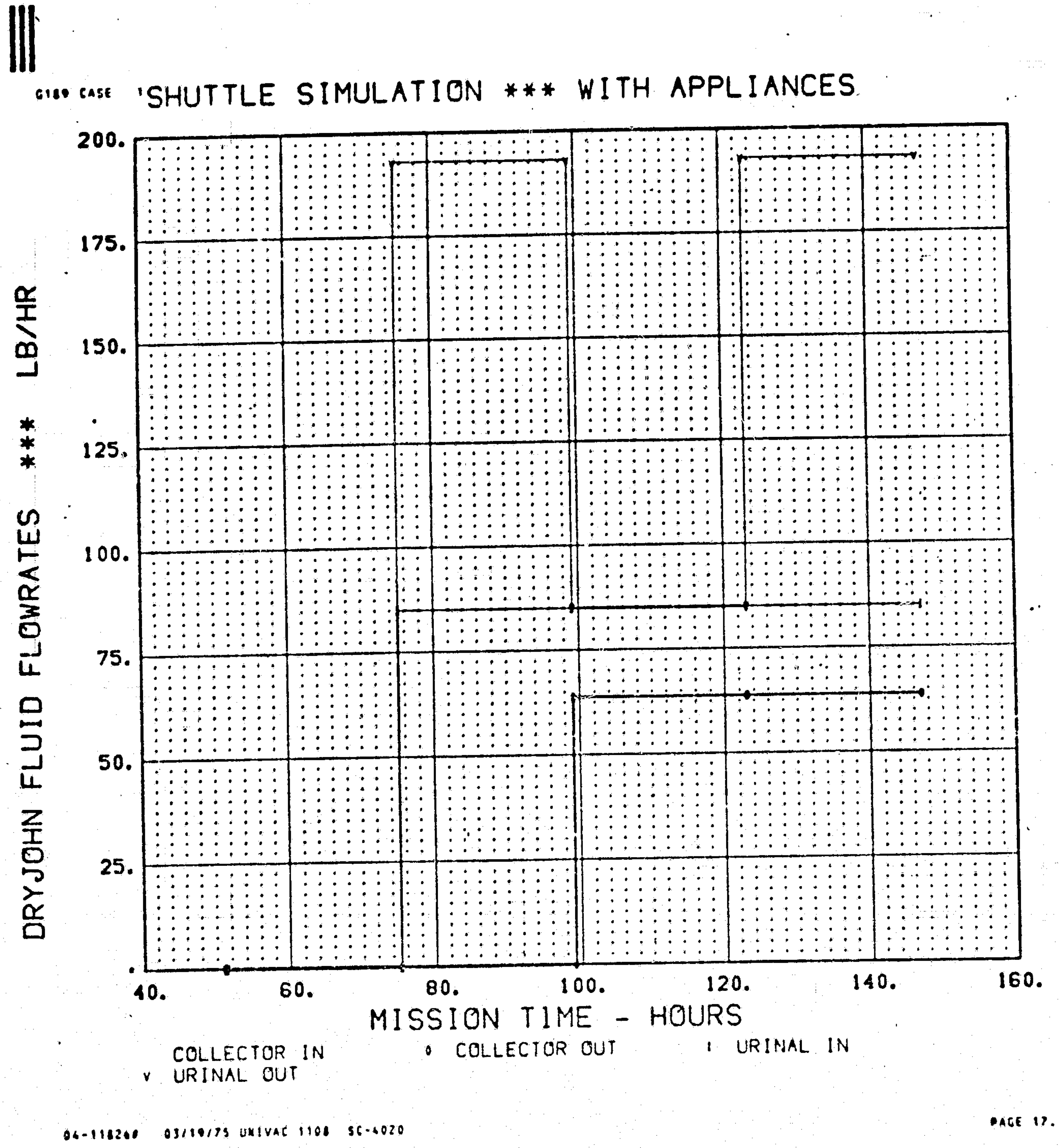

Figure 5-7. Dryjohn Fluid Flow Rates in Shuttle Appliances Simulation

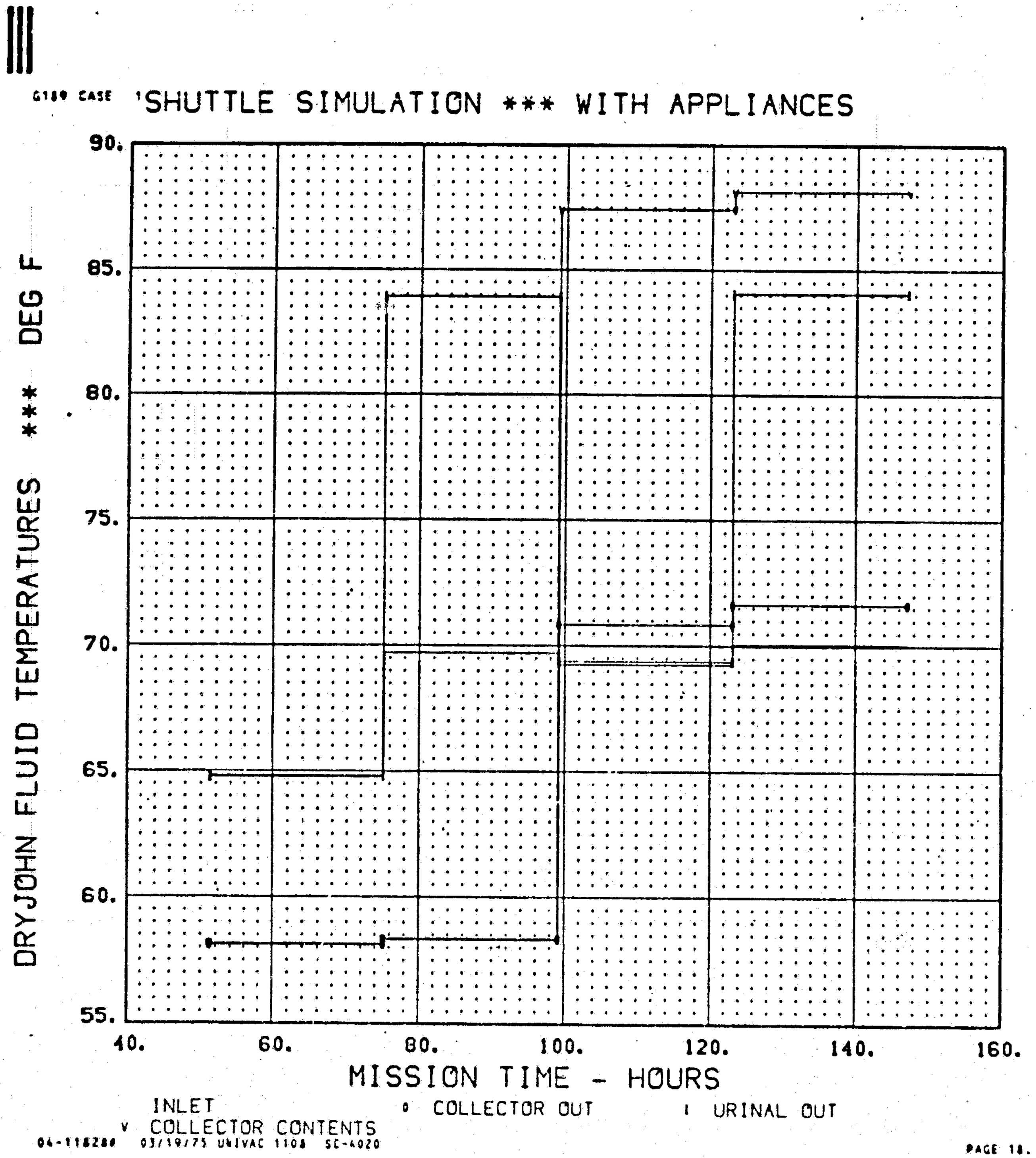

Figure 5-8. Dryjohn Temperatures in Shuttle Appliances Simulation

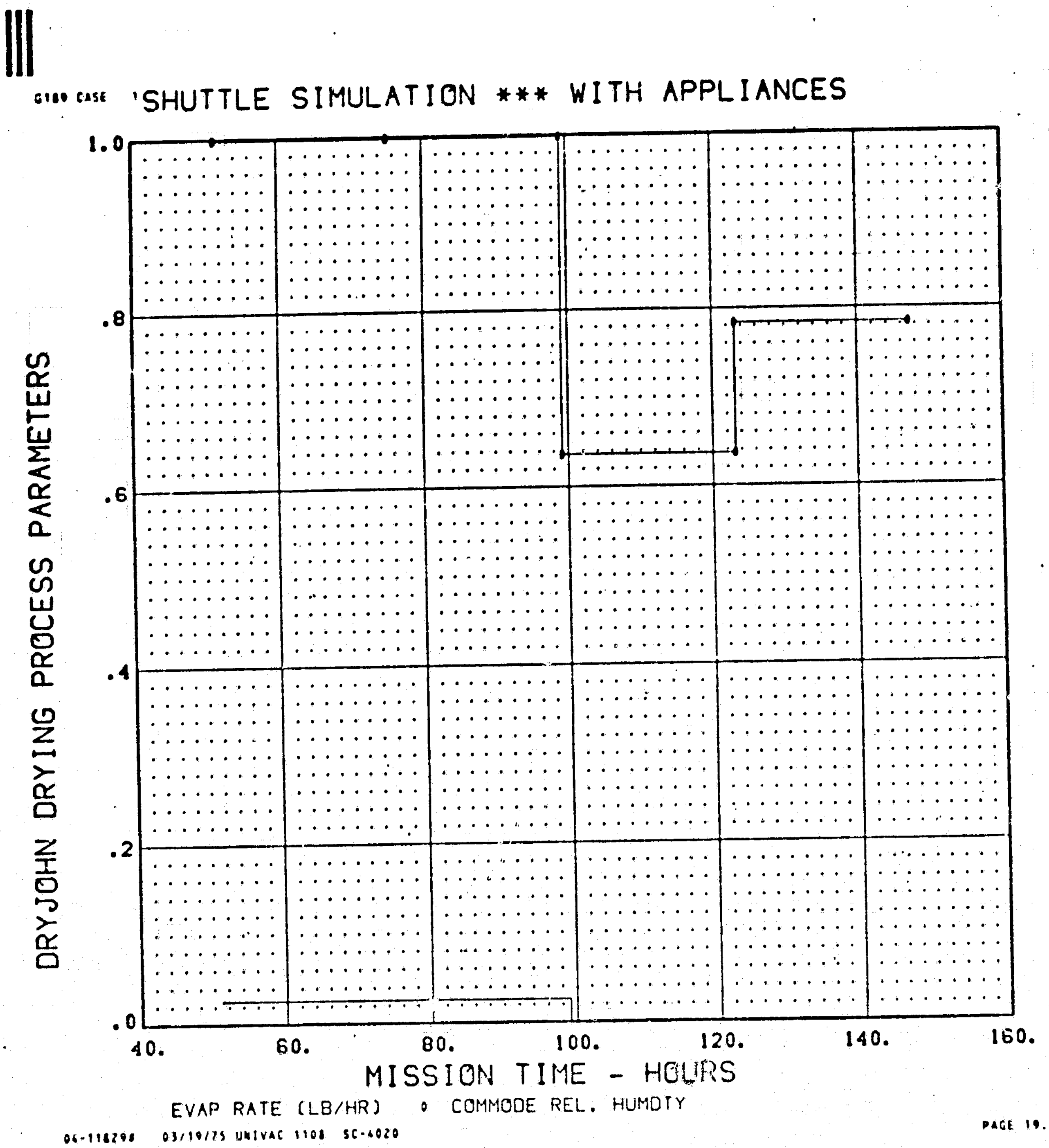

Figure 5-9. Dryjohn Collector Evaporation Parameters in Shuttle Appliances Simulation

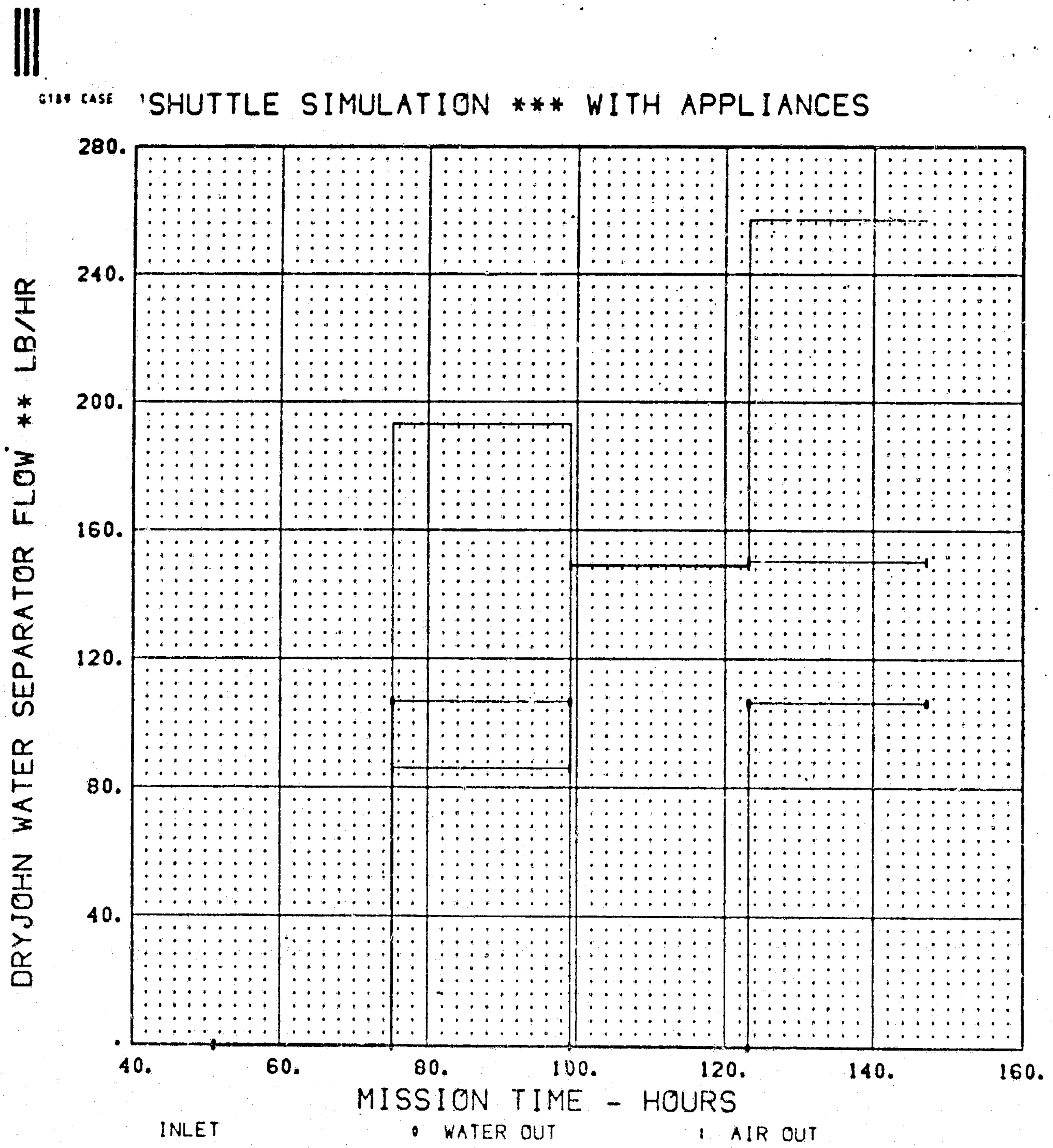

Figure 5-10. Dryjohn Water Separator Performance in Shuttle Appliances Simulation

6.0 SPACE STATION APPLIANCES SIMULATION

To demonstrate that the new appliance subroutines are operational in an all-up G-189A system simulation, they were run in a Space Shuttle Orbiter and Modular Space Station G-189A ECLSS model. The appliances included in the models were taken from trade studies of many candidate concepts (Reference 1) and represent the optimum set of appliances for each vehicle. The results of these trade studies for a 180-day, six-man Modular Space Station are shown in Table 2-2.

6.1 GENERAL SPACE STATION MODEL DESCRIPTION

Since no operational model of the Space Station ECLSS was found prior to this appliance simulation effort, a simplified model of the pertinent subsystems was developed in which to check the appliance subroutines. The Modular Space Station concept (Reference 29) involves eight separate compartments for crew habitability, work, experiments, etc. Only two of these compartments were included in the model. The first includes most of the personal hygiene equipment, and the other the crew eating and sleeping quarters. The G-189A flow schematic for these cabins and associated appliances is shown in Figure 6-1. The water loop used to supply these appliances is shown in Figure 6-2. Flow schematics for the other appliances not included in these two flow loops are shown in Figure 6-3. The following appliances were included in the Space Station model:

- Refrigerators
- Freezers
- Food trays
- Reverse osmosis unit
- Shower
- Clothes washer/dryer
- Dishwasher/dryer
- Dryjohn
- Wet wipe wetting unit

C-4

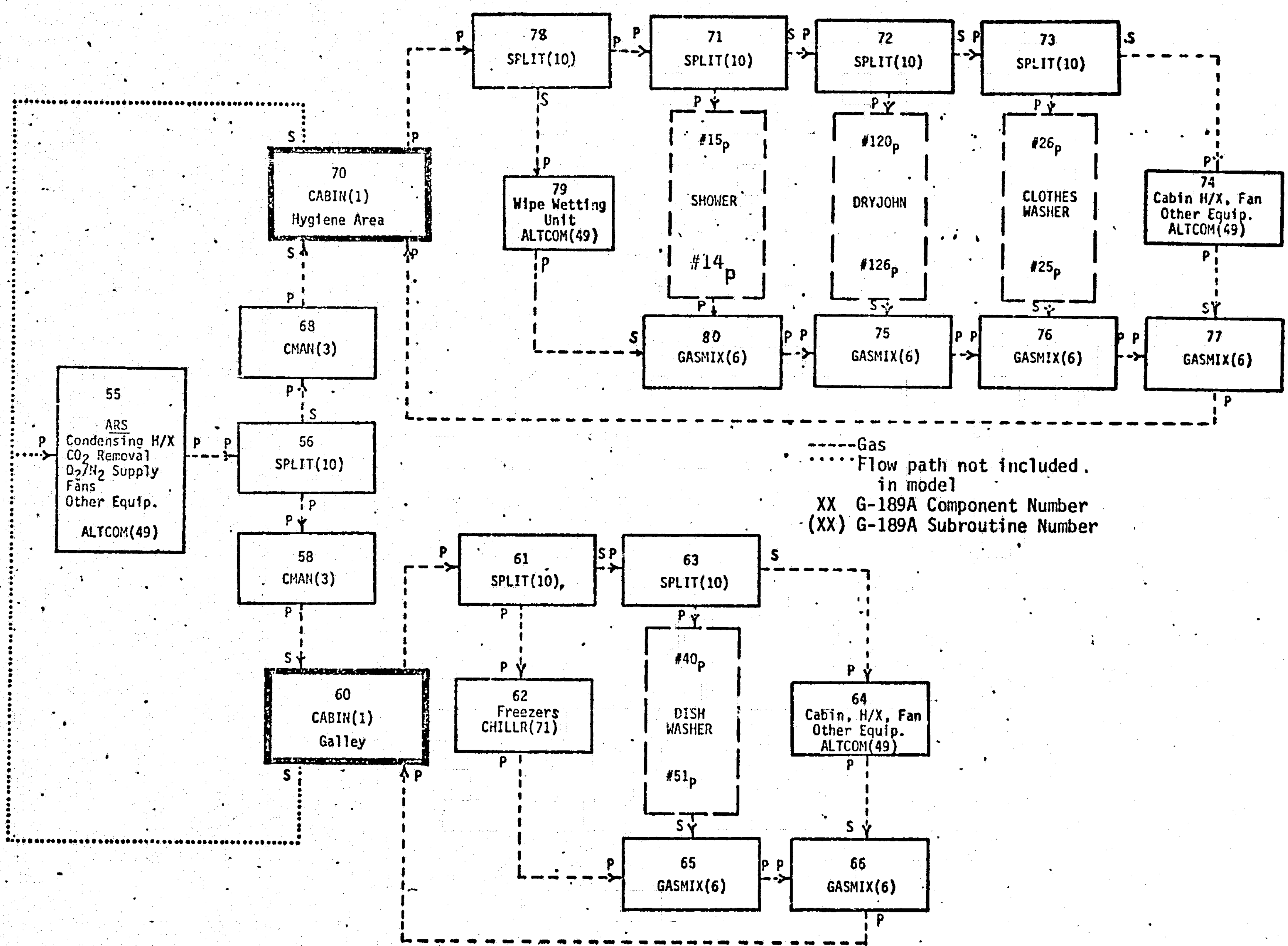

Figure 6-1. G-189A Flow Schematic of Space Station Cabin Gas Loop

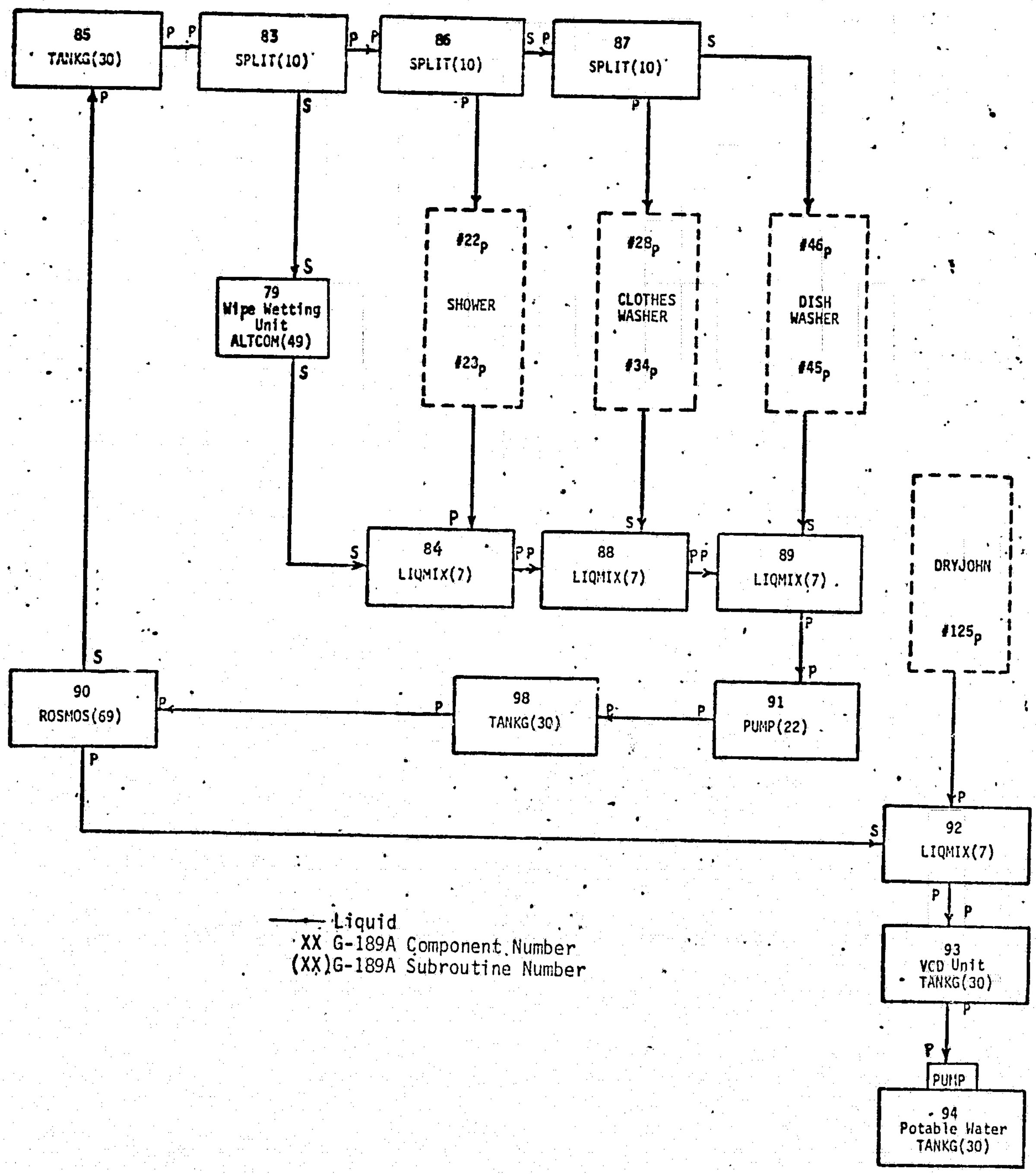

Figure 6-2. G-189A Flow Schematic of Space Station Water Loop

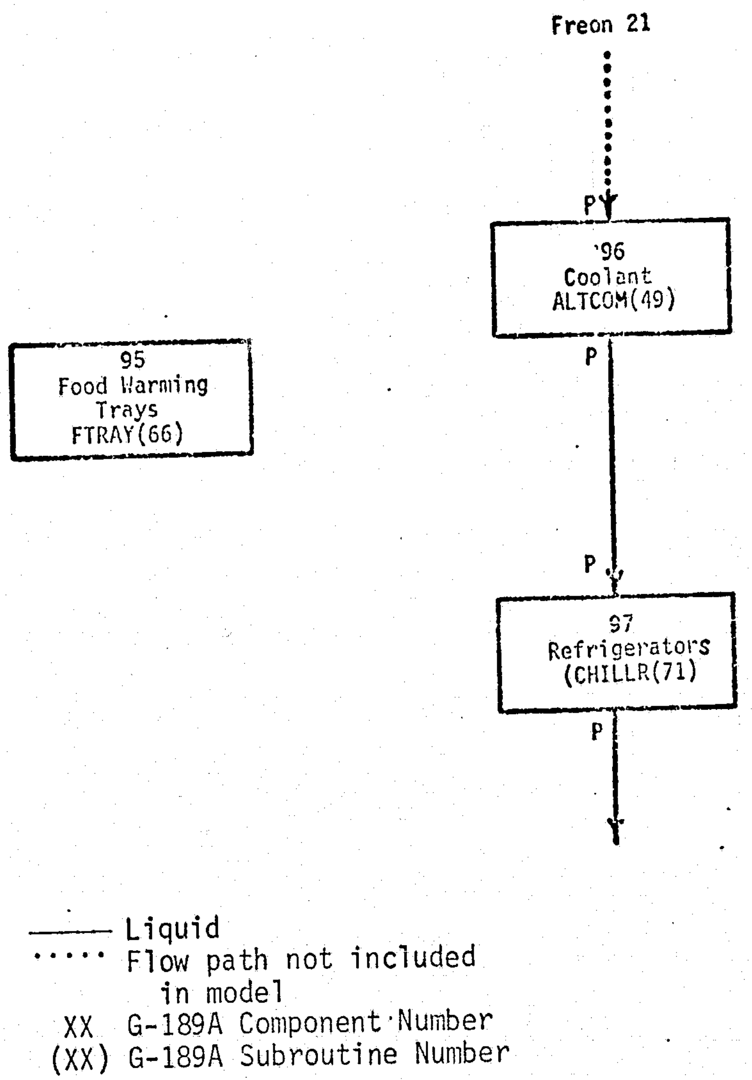

Figure 6-3. Other G-189A Components Used in Space Station Model

6.1 (Continued)

A description of each appliance is given in the following paragraphs. The appliances were run in a 10-hour transient simulation according to a typical daily schedule shown in Figure 6-4. The G-189A input model data and GPOLY subroutines are listed in Appendix F.

6.2 REFRIGERATORS

The CHILLR subroutine was used to model the Space Station refrigerator, component number 97. The input model data used for this component are included in Appendix F. It is identical to the refrigerator model discussed in Paragraph 4.1.2. It was assumed the refrigerator could be connected into the Freon 21 radiator cooling loop. Since this loop was not required anywhere else in the model, a dummy input was made (as shown in Figure 6-3) to define constant Freon inlet flow and properties to the refrigerator. The appliance concept trade study (Reference 1) specified three refrigerators of equal size for Space Station. This was accomplished in a simplified manner for demonstration purposes by assuming all three refrigerator units were identical. The inlet Freon flow to the three refrigerators was multiplied by 1/3 immediately before solution of the component. Thus, only one unit was included directly in the flow loop. After component solution of the single unit, the outlet flow rate was then multiplied by 3 to include the total effect of all three refrigerator units. The heat loss from the refrigerators by door opening and conduction through the walls [in R(53)] was also multiplied by 3 and added to the total galley cabin sensible heat load (in R-array location 66) in the GPOLY1 subroutine. The locker door was assumed to be opened once every hour, as seen in the schedule of Figure 6-4.

6.3 FREEZERS

The CHILLR subroutine was used to model the Space Station freezer, component number 62. The input model data used for this component are included in Appendix F. It is identical to the freezer model discussed in Paragraph 4.1.4.

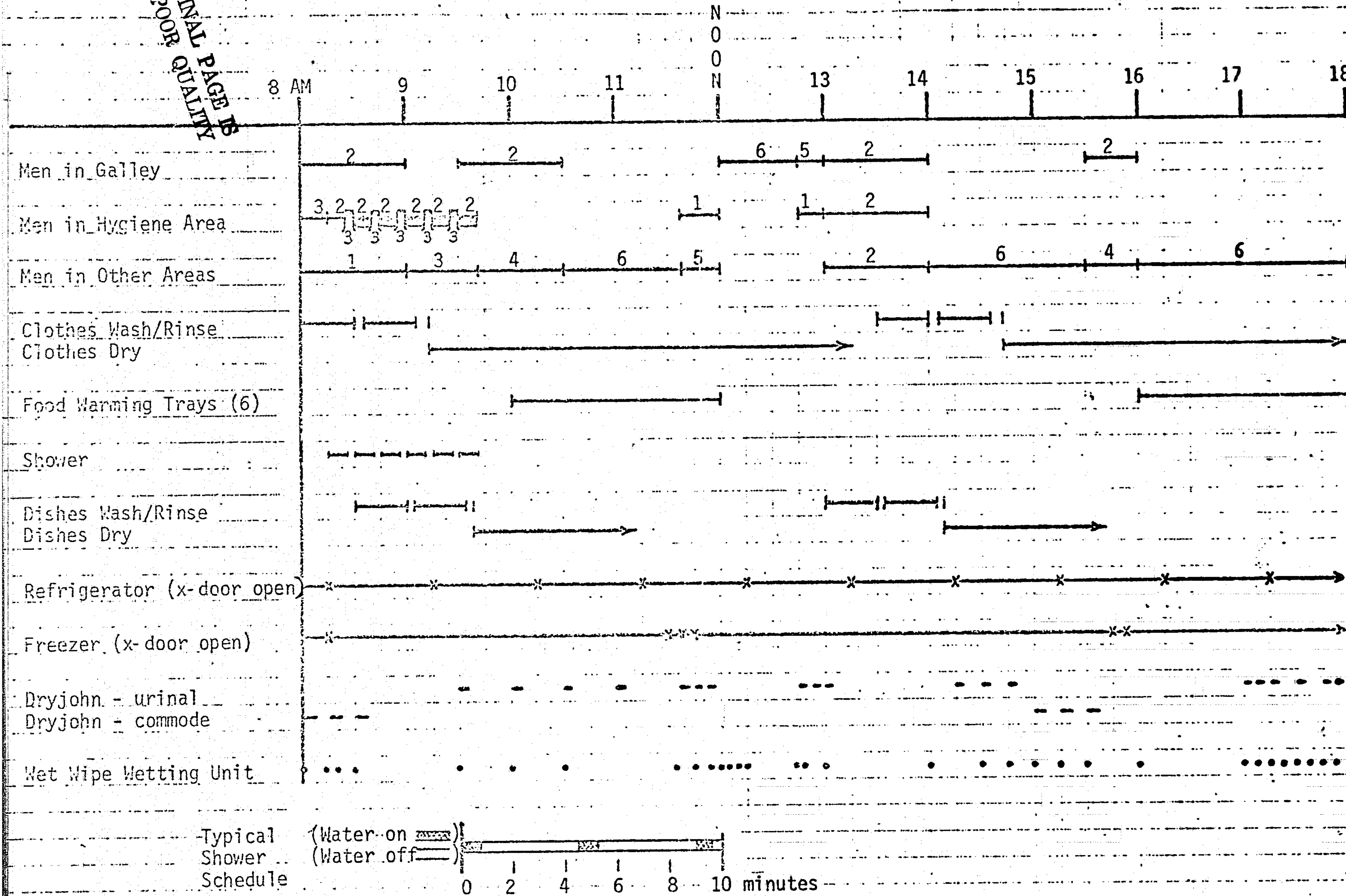

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Figure 6-4. Appliance Usage Schedules in Space Station Transient Simulation

6.3 (Continued)

A self-contained vapor compression refrigeration unit once under consideration for use on Shuttle (Reference 22) was assumed to provide the cooling. The freezer was included in the cabin gas loop, Figure 6-1, to provide cabin air to cool the condenser.

The appliance concept trade study (Reference 1) specified 13 freezers of equal size for Space Station. This was accomplished in the model in the same manner as was described in Paragraph 6.2 for the refrigerators; that is, all 13 units were assumed to have the same thermal performance, so a solution was actually obtained for only one. The air flow to the 13 freezer units was therefore divided by 13 in GPOLY1 immediately before component solution and then multiplied again by 13 immediately after in GPOLY2. The heat loss from the freezer by door opening and conduction through the walls was also multiplied by 13 and added to the total galley cabin sensible heat load (in R-array location 66) in GPOLY1. Thus, the full effect of all 13 freezer units was included in the Space Station model. The freezer locker doors were assumed to be opened at random intervals during the day according to the schedule in Figure 6-4.

6.4 FOOD HEATING/SERVING TRAYS

The FTRAY subroutine was used to model the Space Station food heating trays, component number 95. The input model data for this concept are included in Appendix F. It is identical to the food tray model discussed in Paragraph 4.2. The heating trays were not connected directly into any fluid flow loop, as indicated in Figure 6-3. The heat dissipated to the cabin from the trays (stored in the food tray R-array location 53) was added in GPOLY1 to the total cabin heat load in galley cabin component R-array location 66. The trays are warmed for 2 hours just before the noon and evening meals, as seen in the schedule in Figure 6-4. It was assumed the trays for all six crewmembers were heated at the same time; thus, only a single component was used to simulate all six trays. It was assumed each tray had three heating cavities being warmed. Thus, the subroutine internally multiplies the heat loss for one heated cavity by 18 to account for all six trays.

6.5 REVERSE OSMOSIS UNIT

The ROSMOS subroutine was used to model the Space Station reverse osmosis unit, component number 90, as seen in Figure 6-2. The input model data used for the component are included in Appendix F. The model design is identical to the unit described in Paragraph 4.3 except that it operated near ambient temperature, so no thermal conduction path was included between the unit and its surroundings. The product water was used as wash water for the clothes washer, dishwasher, shower, and wipe wetting unit.

6.6 SHOWER

The Space Station whole body shower interfaces with the cabin gas loop, Figure 6-1, and the water loop, Figure 6-2, for the required air and water flows. The SHOWER subroutine was used to model the shower stall, with the associated peripheral equipment (heaters, fan, pump, etc.) simulated with standard G-189A component subroutines. The shower stall model data, component 3, are identical to the unit described in Paragraph 4.4. The flow schematic for the complete shower unit is shown in Figure 6-5. A description of the shower subsystem is given in References 19 and 24. The shower is operated six times in the morning for 10 minutes each, as shown in the schedule in Figure 6-4. The input data used to model each component are included in Appendix F.

An electrical air heater was assumed (component number 19), with a thermostatically controlled temperature of 105°F. A heat exchanger was used to warm the inlet water. However, rather than model the heat exchanger, a dummy component (number 22) was used with a fixed outlet water temperature of 105°F. A vortex water separator was used which does not require a drive motor. This component is treated using GPOLY2 logic (shown in Appendix F, GPOLY2 statement number 2000) which separates the inlet air and entrained water into two outlet flow streams: one gas, the other liquid. Since the air and water were mixed in the shower stall, no further mixing is required in this component. A fan and pump component (numbers 16 and 23) were used to circulate the air and water.

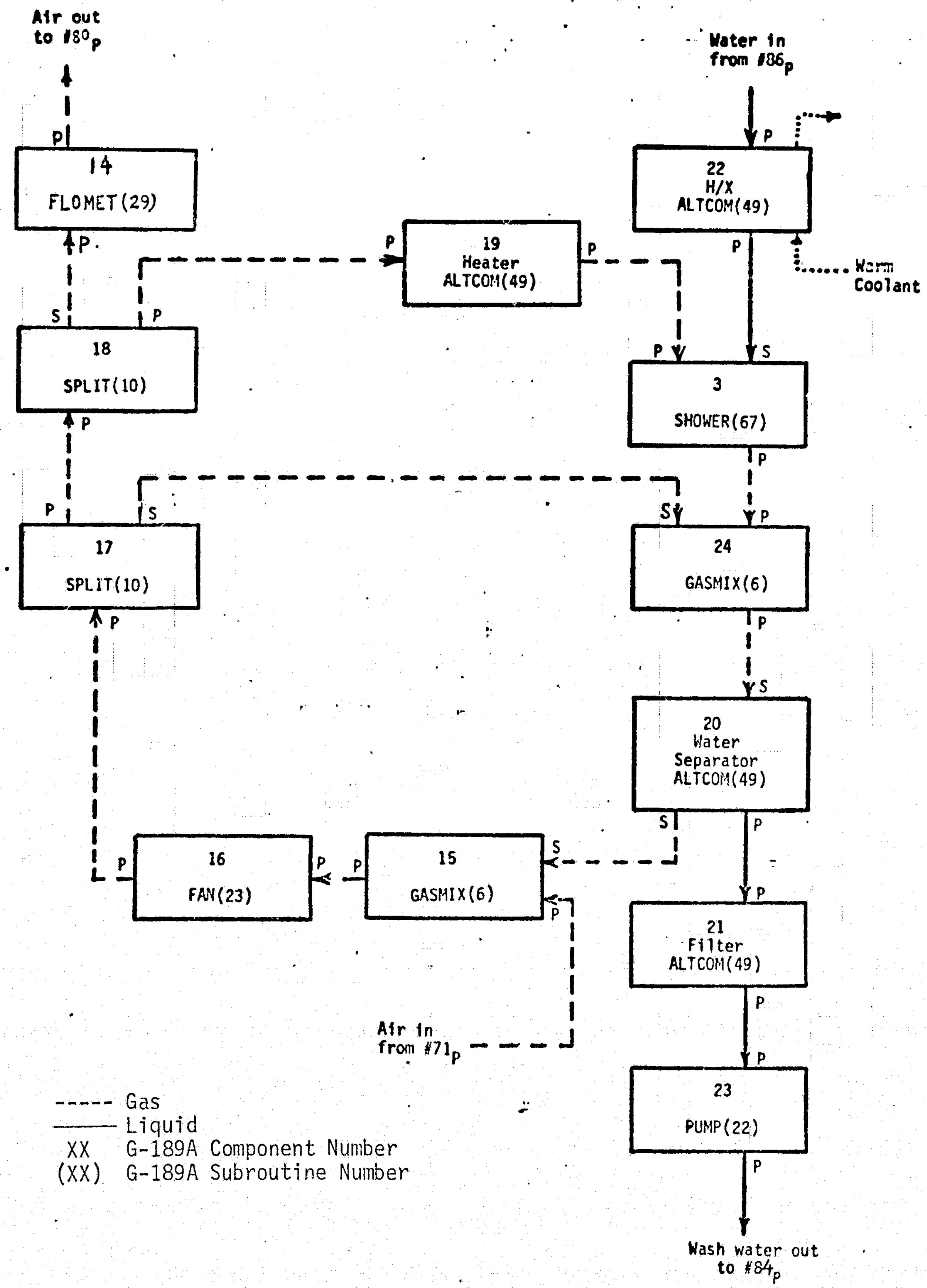

Figure 6-5. G-189A Flow Schematic of Space Station Shower Model

The normal order of solution for the shower loop components in Figure 6-5 is as follows:

$$22 \rightarrow 15 \rightarrow 16 \rightarrow 17 \rightarrow 18 \rightarrow 19 \rightarrow 3 \rightarrow 24 \rightarrow 20 \rightarrow 21 \rightarrow 14 \rightarrow 23$$

When the shower is not being used but the stall is at an average temperature above ambient due to previous usage, the normal solution path shown above is followed to permit thermal exchange between the stall and cabin environment. When the shower is not in use and the stall is in thermal equilibrium with the cabin, the solution of the loop is skipped over (solution path 22 → 23) to conserve computer time.

When the shower subsystem is being used, the 90-second system computing time step was too large to achieve satisfactory accuracy of the shower loop mass balance. Instead, a 30-second interval was found to be adequate for solving the shower loop, and three successive solutions were obtained each system time step to achieve a total of 90 seconds. For each 30-second time step, an iterative solution was used to achieve a mass balance within the loop. To begin the iterations, an estimate of the outlet flow conditions from component 20 is first made, after which the loop is solved in the normal sequence given previously through component 20. The computed outlet conditions from component 20 are then compared with their estimated values. If different, an updated estimate of the component 20 outlet flow is made, and the solution of the shower loop repeated, beginning with component 15. These iterations are continued until the estimate of the component 20 outlet flow conditions agrees with the actual computed values within 0.1 percent. The logic required to accomplish this iterative procedure is included in the GPOLY2 listing, Appendix F, under statement numbers 2000, 2200, and 2300 (corresponding to components 20, 22 and 23 respectively). Usually only 3 or 4 iterations are required to achieve convergence. These iterations assure that the overall mass balance within the shower subsystem (net mass inlet equal to net mass outlet) is valid. Since the shower is usually activated for only relatively short periods compared with other events (e.g., a maximum of six 10-minute showers per day for Space Station), the required computer execution time for these added iterations is felt to be negligible.

6.7 CLOTHES WASHER/DRYER

The Space Station clothes washer/dryer interfaces with the cabin gas loop, Figure 6-1, and the water loop, Figure 6-2, for the required air and water flows. The WASDRY subroutine was used to model the washer/dryer drum. The input model data for this component are identical to the washer/dryer model described in Paragraph 4.5. The associated peripheral equipment (heaters, fan, pump, etc.) were simulated with standard G-189A component subroutines. No washer/dryer unit has currently been built for space use. However, preliminary design effort has been made, as described in Reference 21, from which much of the Space Station washer/dryer design details have been taken. The flow schematic for the complete clothes washer/dryer unit is shown in Figure 6-6. The input data used to model each component are included in Appendix F. The washer/dryer is operated twice per day, as shown in the schedule in Figure 6-4.

Electrical heaters were used to warm the inlet air and water. The water heater (component number 29) was thermostatically controlled to 155°F. The air heater (component number 26) has one setting of 70.34 watts (240 Btu/hr); it turns on when the dryer outlet air is below 135°F, and off when the air is above 155°F. A fan and pump (components 25 and 32) were used to circulate the air and water. Since the washer drum rotates, it was assumed the drum can be designed to act as a water separator: removing water at the outer periphery of the housing, and removing air near the center of tub rotation. Thus, no water separator was included in the flow schematic. An accumulator tank was used (component number 35) to store rinse water to subsequently be used as wash water. The tank dissipates heat to the cabin air through the tank walls. This heat is stored in R-array location 53 and is added in GPOLY1 to the total cabin heat load in cabin component R-array location 66. Likewise, the clothes washer heat lost through its structure (in R-array location 53) is also added to the total cabin heat in R(66). The washer is operated in all seven cycle phases described in Paragraph 3.5, with 30 minutes allowed for washing and 30 minutes for rinsing. The automatic cycle phase switching logic, described in Paragraph 3.5, was used to control the washer/dryer operation. The drying phase was assumed to terminate when the residual water retention in the clothes material was 0.05 lb water/lb clothes.

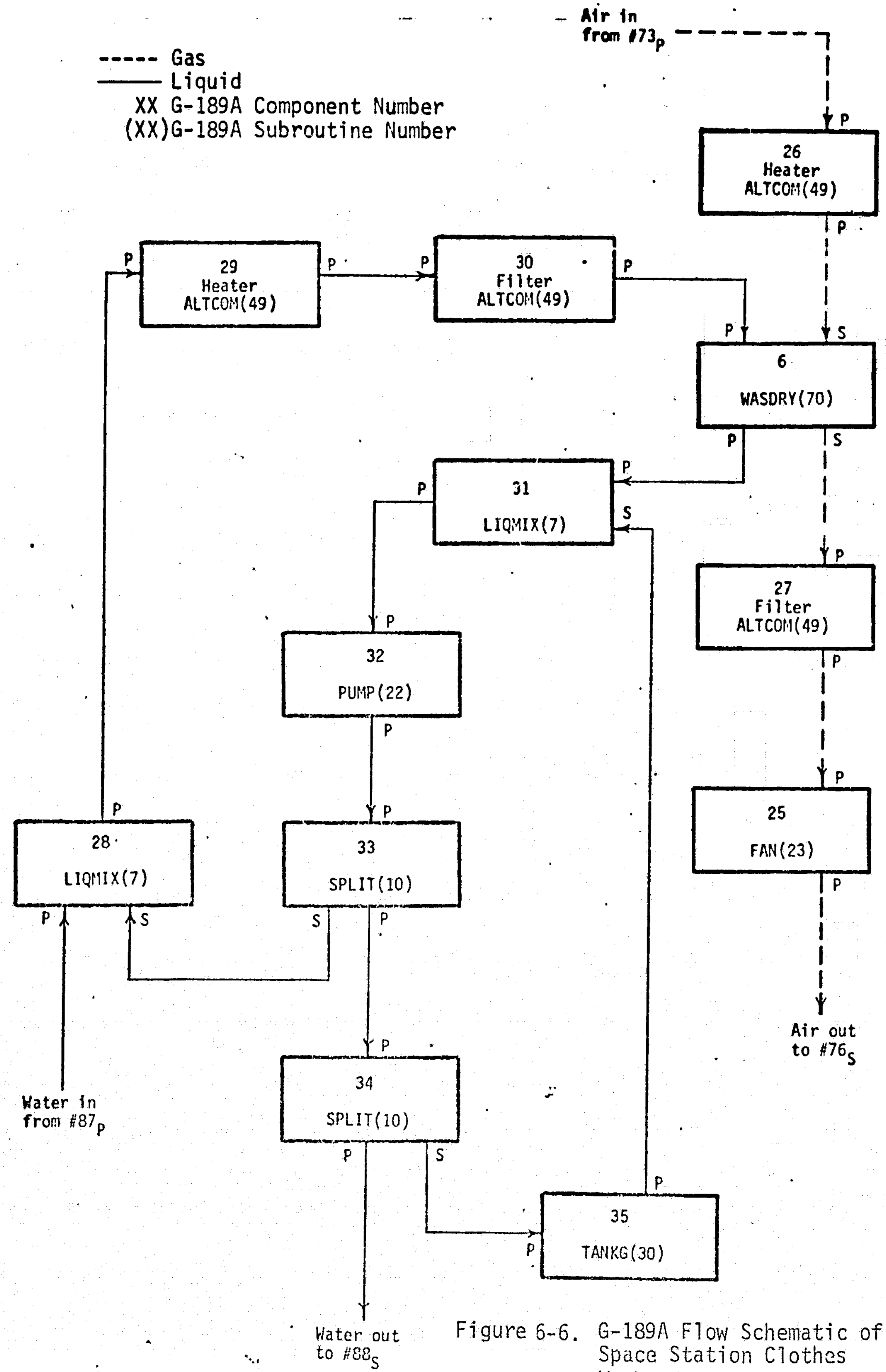

Figure 6-6. G-189A Flow Schematic of Space Station Clothes Washer/Dryer Model

6.8 DISHWASHER/DRYER

The Space Station dishwasher/dryer interfaces with the cabin gas loop, Figure 6-1, and the water loop, Figure 6-2, for the required air and water flows. The WASDRY subroutine was used to model the washer/dryer drum. The input model data for this component are identical to the washer/dryer model described in Paragraph 4.5. The associated peripheral equipment (heaters, fan, pump, etc.) were simulated with standard G-189A component subroutines. No washer/dryer unit has currently been built for space use. However, studies of various concepts have been made, Reference 27, from which some of the Space Station washer/dryer design details have been taken. The flow schematic for the complete dishwasher/dryer unit is shown in Figure 6-7. The input data used to model each component are included in Appendix F. The washer/dryer is operated three times per day. However, only two of these times occur within the 10 hours of mission time being solved, as shown in the schedule in Figure 6-4.

Electrical heaters were used to warm the inlet air and water. The water heater (component number 47) was thermostatically controlled to 155°F. The air heater (component number 40) has one setting of 70.34 watts (240 Btu/hr); it turns on when the dryer outlet air is below 135°F, and off when the air is above 155°F. A fan and pump (components 43 and 51) were used to circulate the air and water. Since the washer drum rotates, it was assumed the drum can be designed to act as a water separator: removing water at the outer periphery of the housing, and removing air near the center of tub rotation. Thus, no water separator was included in the flow schematic. An accumulator tank was used (component number 41) to store rinse water to subsequently be used as wash water. The tank dissipates heat to the cabin air through the tank walls. This heat is stored in R-array location 53 and is added in GPOLY1 to the total cabin heat load in cabin component R-array location 66. Likewise, the dishwasher heat lost through its structure (in R-array location 53) is also added to the total cabin heat load in R(66). The washer is operated in all seven cycle phases described in Paragraph 3.5, with 30 minutes allowed for washing and 30 minutes for rinsing. The automatic cycle phase switching logic, described in Paragraph 3.5, was used to control the dishwasher/dryer operation.

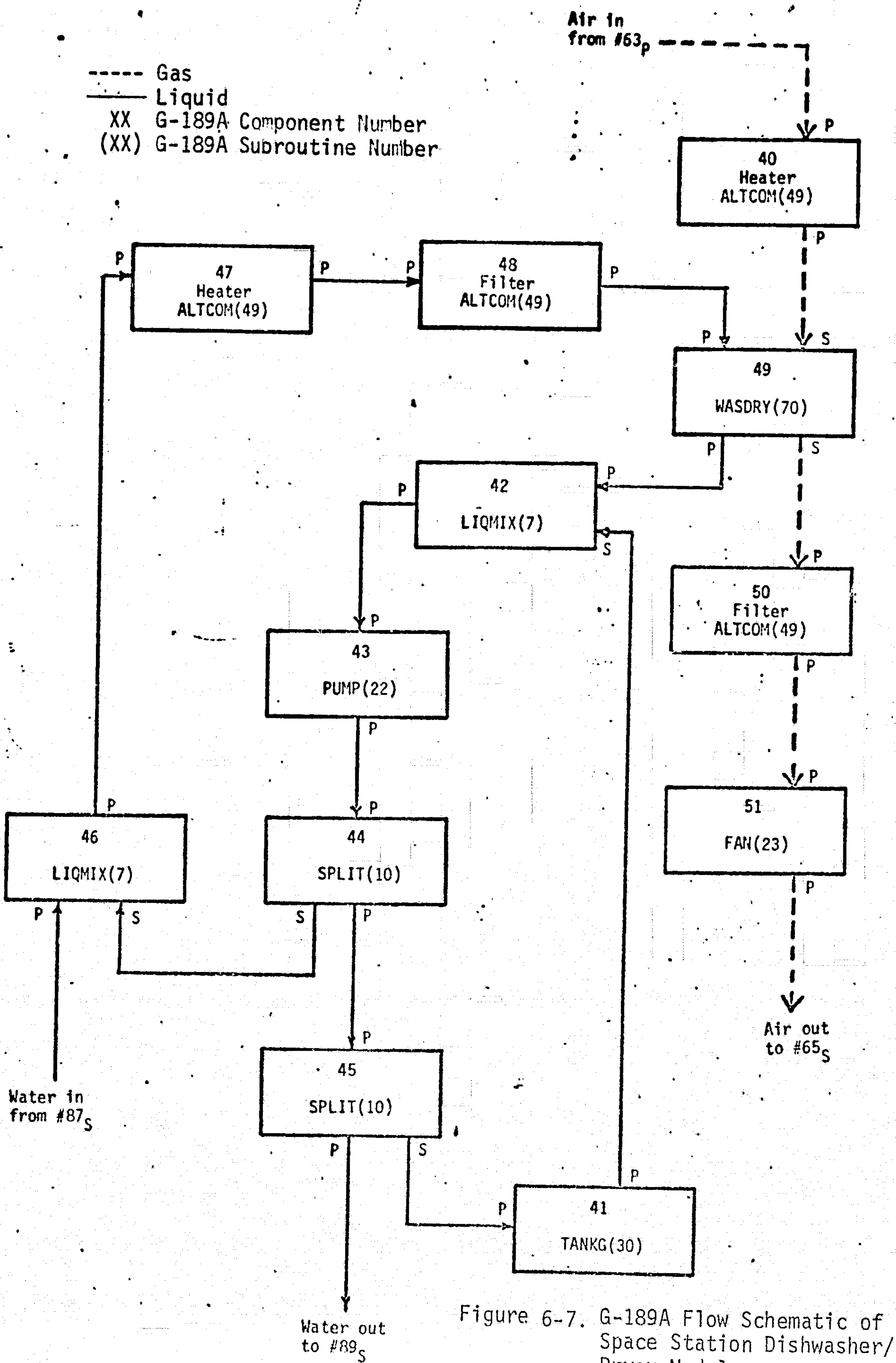

Figure 6-7. G-189A Flow Schematic of Space Station Dishwasher/Dryer Model

6.9 DRYJOHN

The Space Station dryjohn interfaces with the cabin gas loop, Figure 6-1, and the water loop, Figure 6-2. The dryjohn flow schematic is shown in Figure 6-8. The WASTEC subroutine was used to model the urinal and commode collector, component number 121. The input model data for this component are identical to the dryjohn model described in Paragraph 4.6 except for the two minor exceptions noted in that section. The peripheral dryjohn equipment, shown in Figure 6-8, were simulated using standard G-189A components. The G-189A input data used to model each component are included in Appendix F.

The dryjohn was operated as a urinal and commode according to the schedule given in Figure 6-4. When not in operation, or when used as a urinal only, the collector contents were assumed to be evacuated to space vacuum. A motor-driven centrifugal water separator, component 125, was operated at all times when the dryjohn was in use. The logic used to model this separator was included in GPOLY2, statement 12500, and is shown in Appendix F. This involved (1) diverting the entrained liquid to the component primary side, and (2) adding the motor heat to the gas and liquid flows. The water from the separator was directed to the vapor compression-distillation unit, component 93, as shown in Figure 6-2.

6.10 WET WIPE WETTING UNIT

A wipe wetting unit was included in the Space Station model to provide hand washing and partial body washing. This concept, described in Reference 30, dispenses water and soap into a washcloth, then rinses and wrings out the cloth after use. A motor-driven water separator is used with the component to provide inlet air flow and to separate the outlet gas and liquid. This device, component 79, is included in the cabin gas and water loops, Figures 6-1 and 6-2.

The wipe wetting unit is operated intermittently during the day according to the schedule in Figure 6-4 taken from Reference 31. A single use of the unit is assumed to require one computing time step, which was 90 seconds.

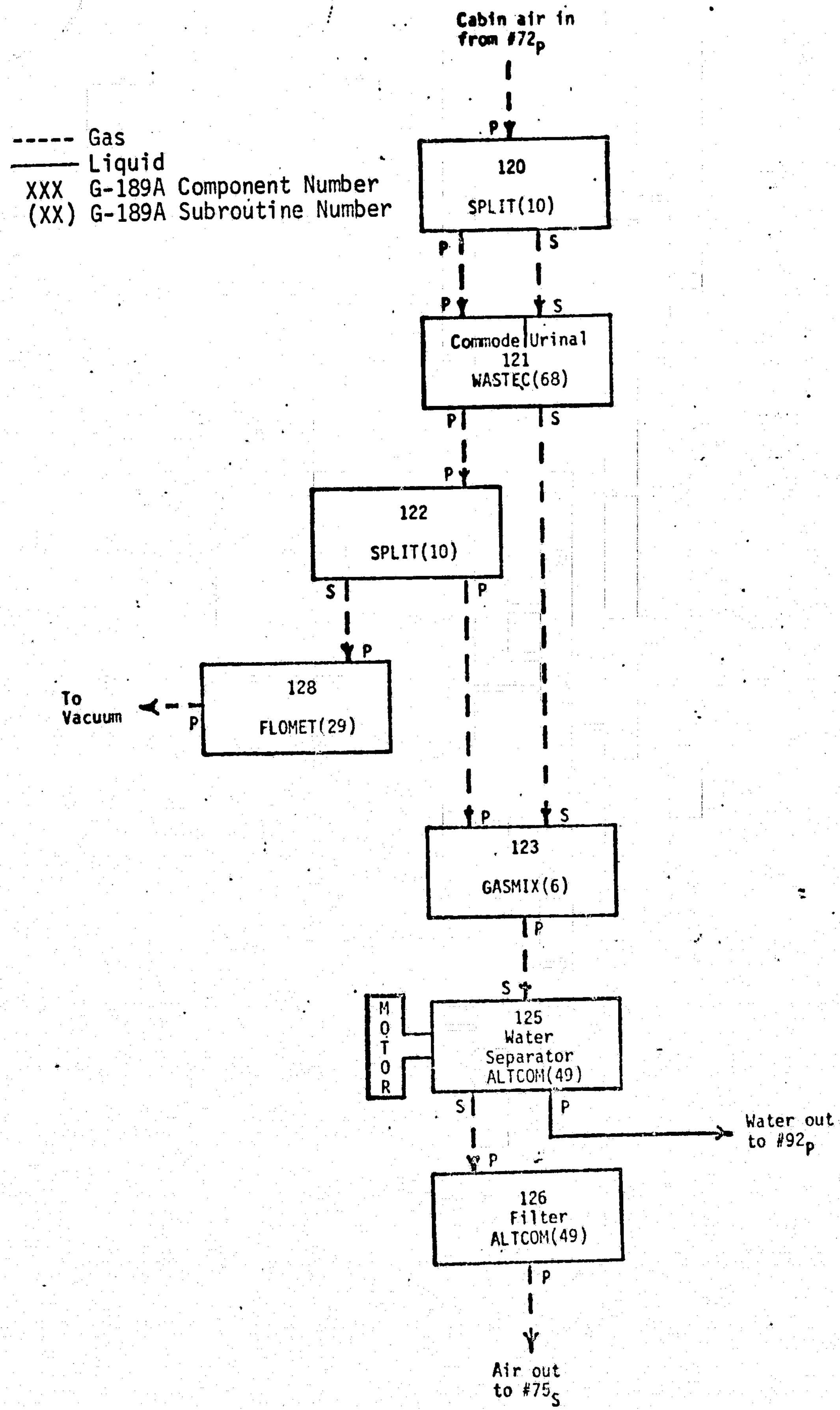

Figure 6-8. G-189A Flow Schematic of Space Station Dryjohn Model

6.10 (Continued)

During this time, the latent and sensible heat specified for this component in Reference 30 are added to the air and water flows. There is 0.00635 gm (0.014 lb) moisture added to the cabin air flow through the device and 84 watts (286.6 Btu/hr) heat added from the separator motor. The air and water are assumed to leave the unit at the same temperature. The calculations required to simulate this operation were included in GPOLY2, statement number 7900, and are given in Appendix F.

6.11 SPACE STATION/APPLIANCES MODEL RESULTS

The Modular Space Station G-189A model described in the preceding sections was run for a 10-hour transient simulation. The appliances included in the model were operated according to the usage schedule shown in Figure 6-4. The results are plotted in Appendix G. These plots show cabin temperatures, humidity, CO_2 levels, and the thermodynamic performance of the appliances. The final solution for the run is also listed in Appendix G in the standard G-189A format.

The total net effect of each appliance subsystem on the Space Station cabin air, coolant system, and water supply is shown in Figures 6-9 through 6-29. For the appliance subsystems requiring more than one G-189A component (clothes washer/dryer, dishwasher/dryer, shower, and dryjohn), the values plotted represent the combined effect from each component. The plotted sensible heat to cabin represents the heat transferred to the cabin air flowing through the unit as well as the structural heat leak from each component in the appliance subsystem. Some of the figures show heating or flow rates as short single-line "spikes" (e.g., Figures 6-9, 6-24, and 6-27). These values represent momentary component operation for one system time step, which was 90 seconds.

The total appliance thermodynamic results shown in Figures 6-9 through 6-29 are shown to illustrate typical appliance operation, and must be interpreted in the context of the specific assumptions used for the Space Station

6.11 (Continued)

demonstration run. For example, the shower water usage and humidifying effect will vary depending on the specific water usage schedule used by the occupant, and the results shown in Figures 6-13 through 6-17 reflect the usage schedule in Figure 6-4. Since the washer/dryer hardware has not yet been designed, reasonable values were assumed for the thermal connections from the frame and accumulators to ambient. These values could vary widely with different structural designs, and this will affect the heat leak to the cabin. The thermal effect from the 13 food freezers could be exaggerated, since only one was modeled directly in the computer run, and the other 12 were assumed identical. Therefore, the cooling units for all 13 freezers cycle on and off together, making a greater cabin heat input (Figure 6-11) than if their operation were staggered.

The net thermodynamic performance of each appliance subsystem is demonstrated in Figures 6-9 through 6-29. The detailed thermodynamic parameters of interest in the individual appliance components are presented in Appendix G. As discussed in Sections 6-2 through 6-10, the appliances included in the Space Station model were of the same design, with the same model input data, as the appliances described in Section 4 for verification of the models. Therefore, the Space Station appliance results plotted in Appendix G may be compared directly with the corresponding results in Section 4. The solutions for each appliance in the two sections are nearly identical, as expected, with all differences attributable to the difference in inlet and ambient conditions. Therefore, these results demonstrate that the new appliance subroutines are valid and operational in an all-up G-189A system model.

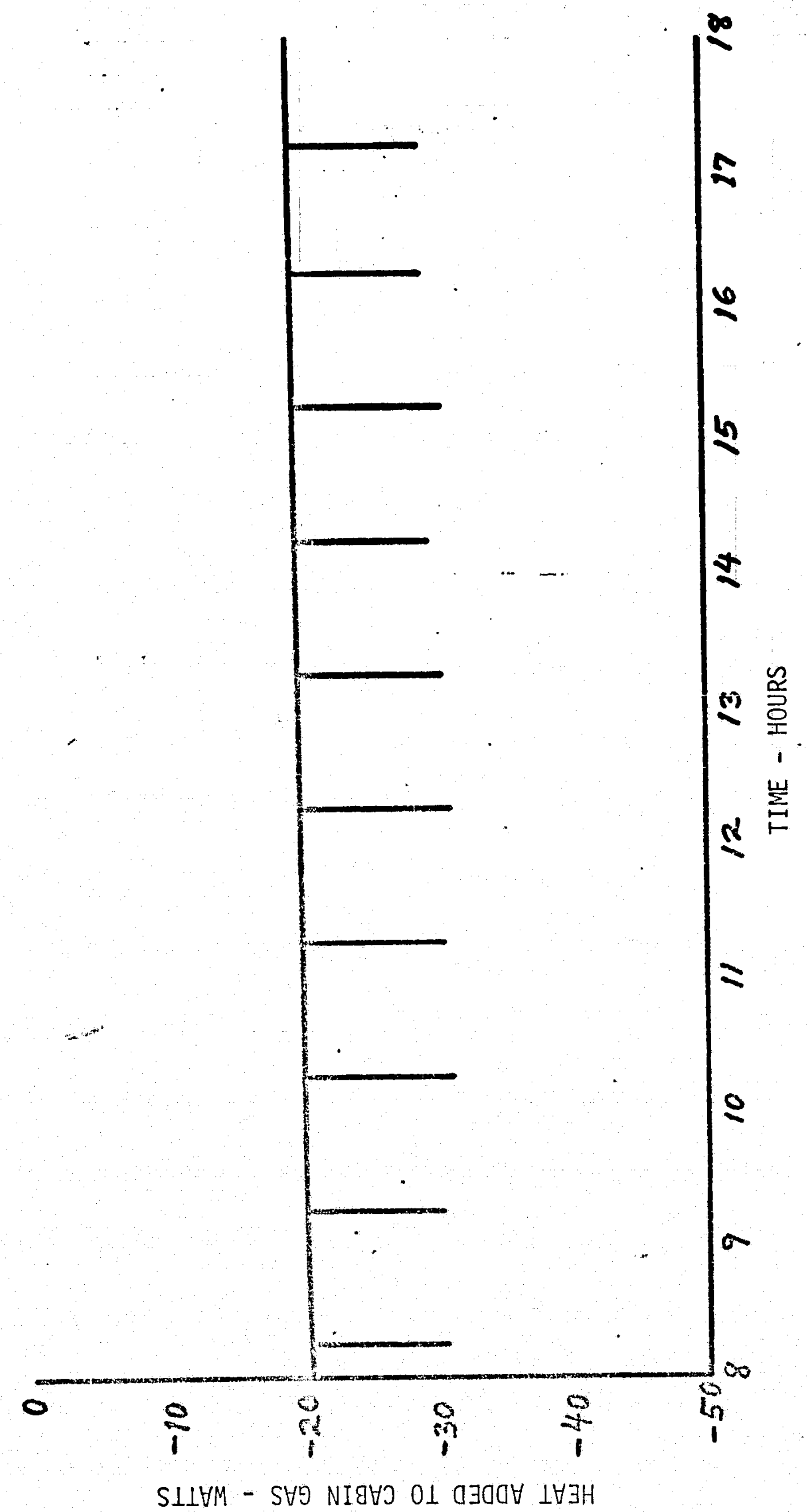

Figure 6-9. Space Station Sensible Heat Input from Refrigerators

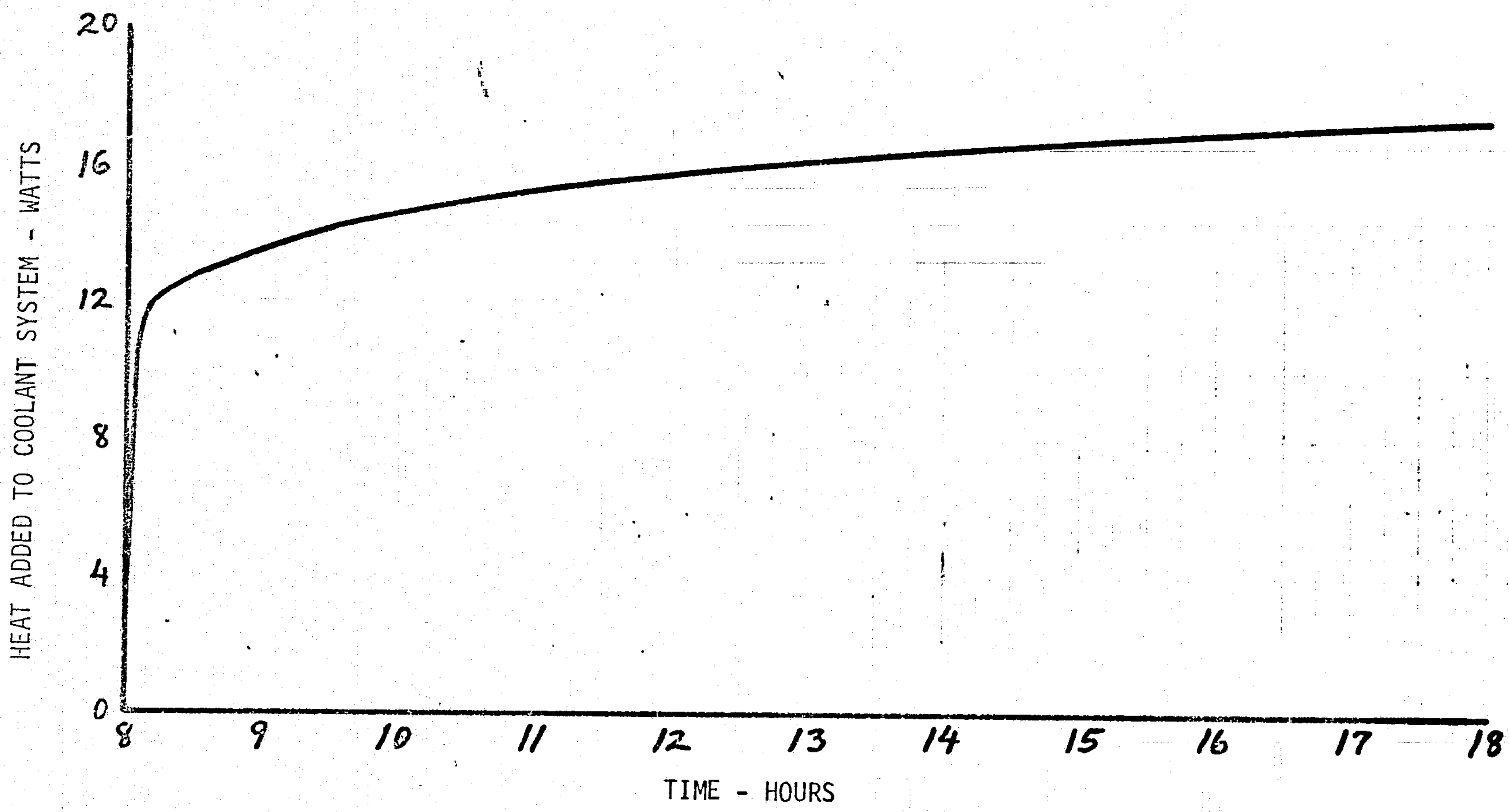

Figure 6-10. Heat Input to Space Station Coolant from Refrigerators

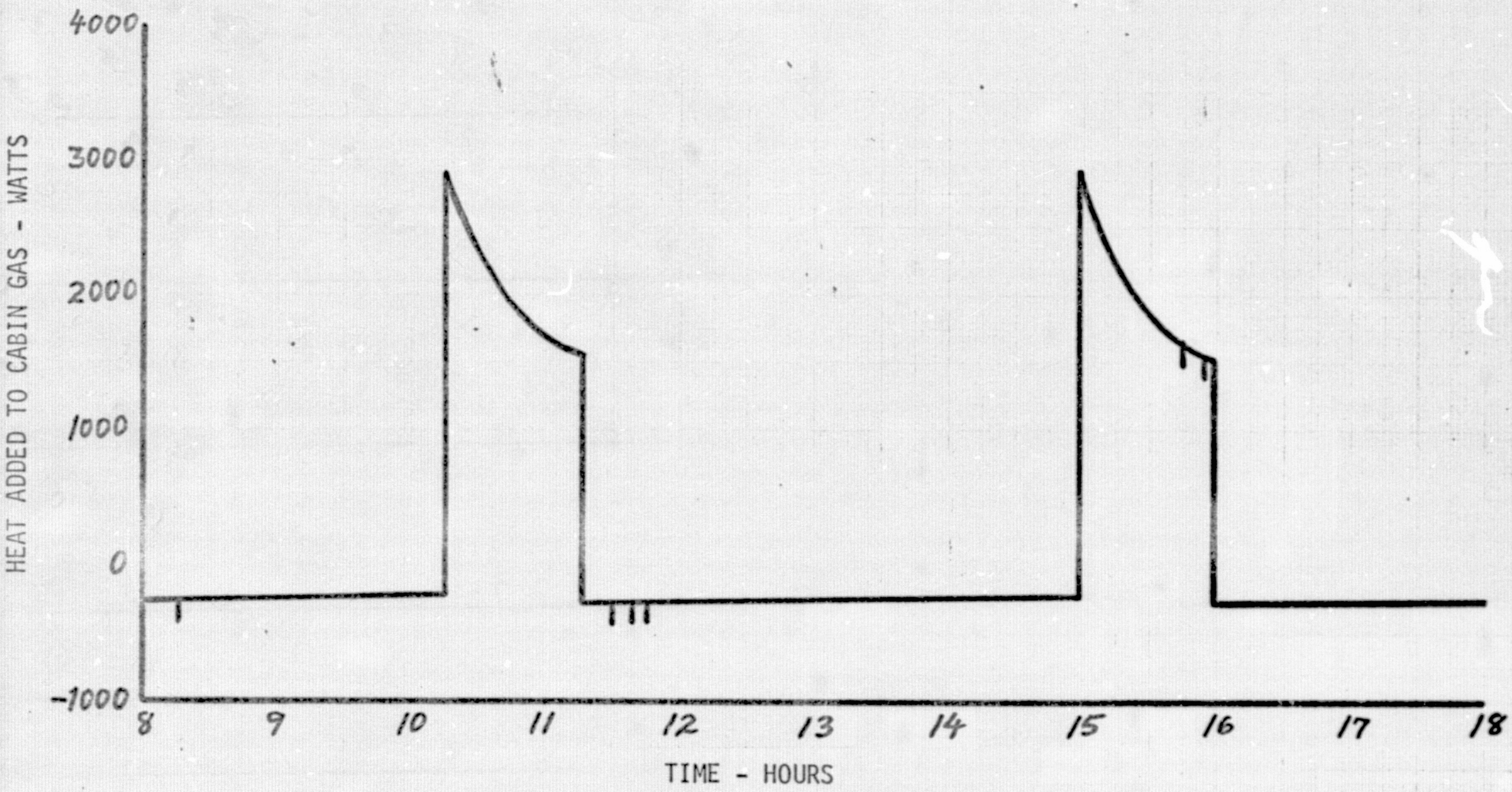

Figure 6-11. Space Station Sensible Heat Input from Freezers

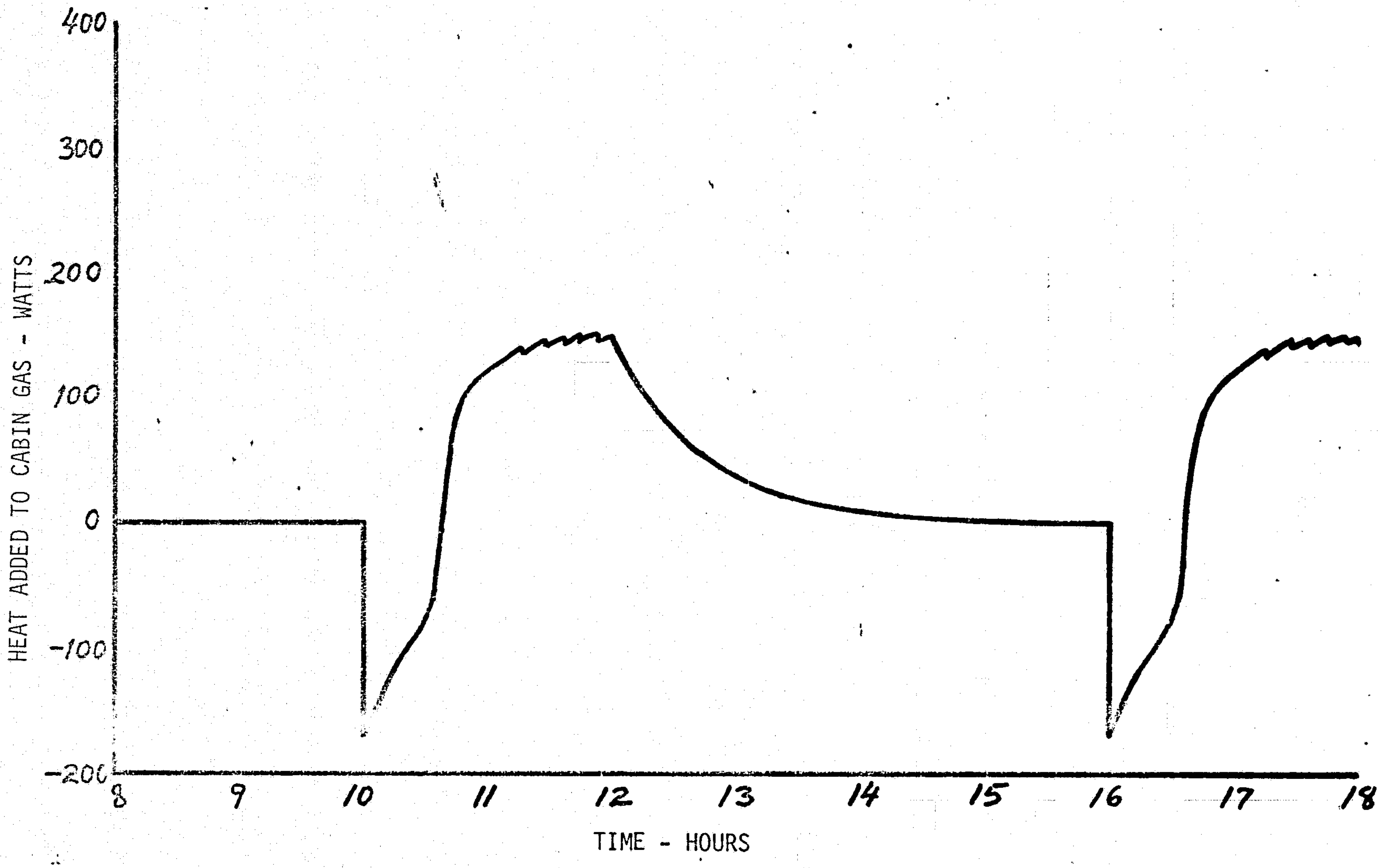

Figure 6-12. Space Station Sensible Heat Input from Food Trays

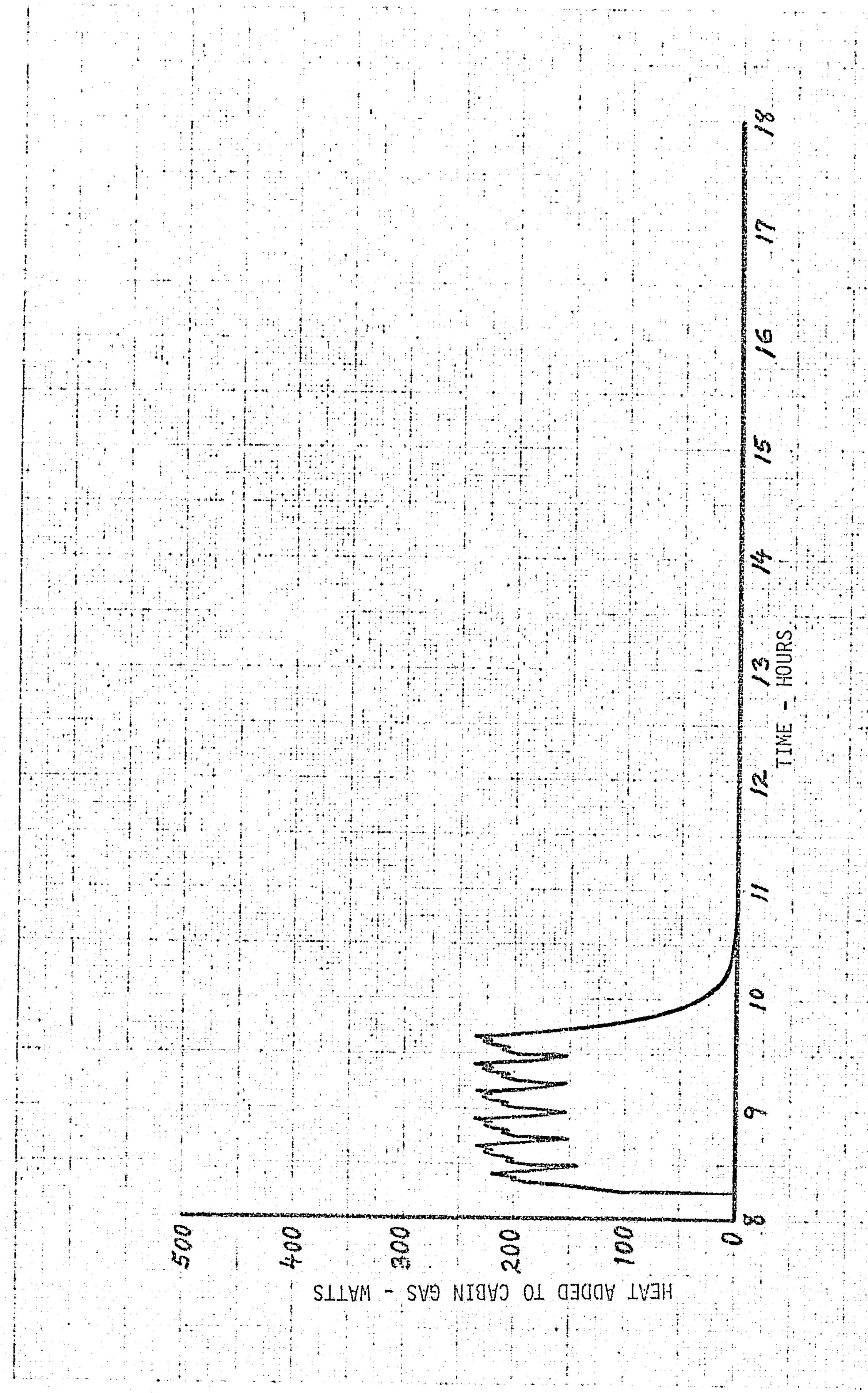

Figure 6-13. Space Station Sensible Heat Input from Shower

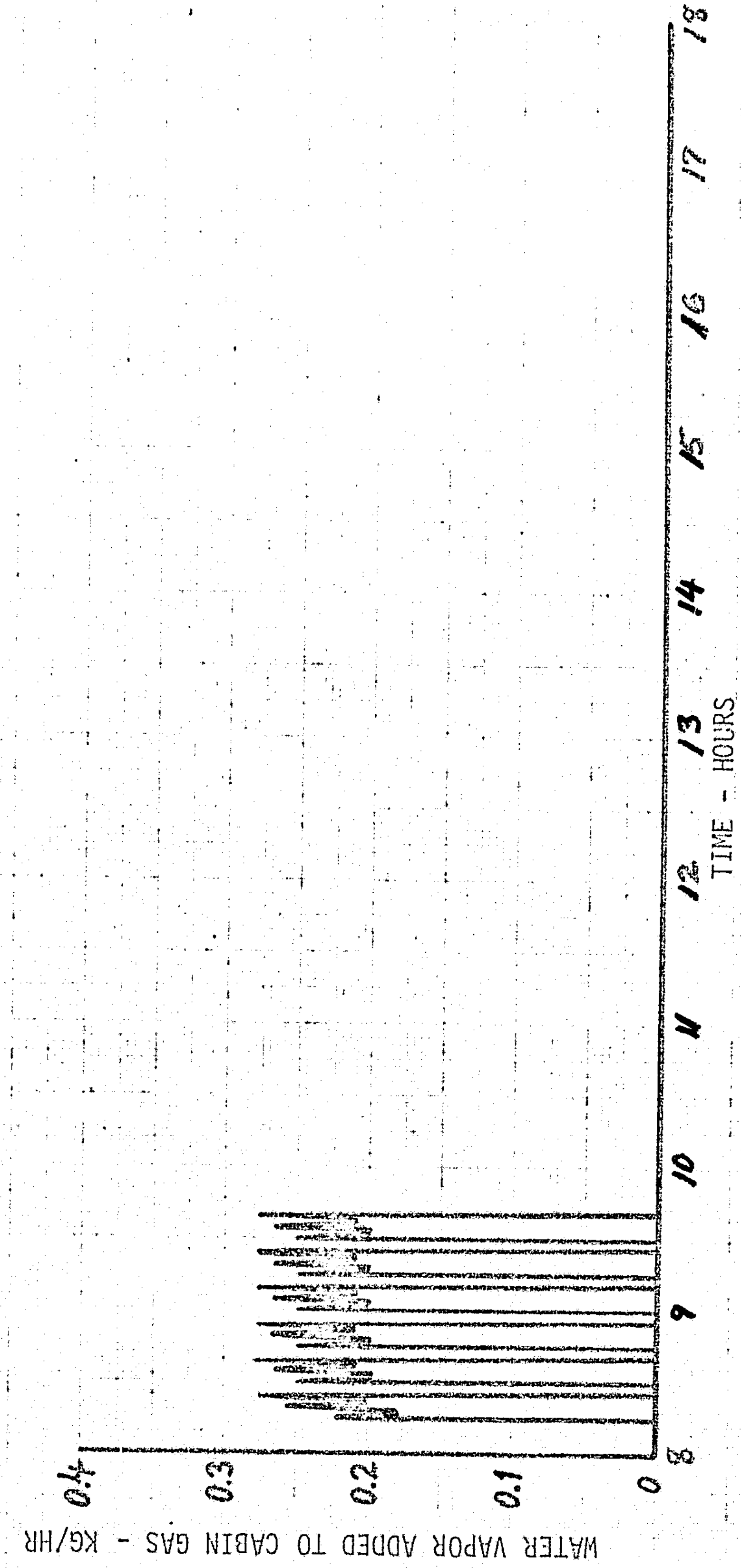

Figure 6-14. Space Station Water Vapor Input from Shower

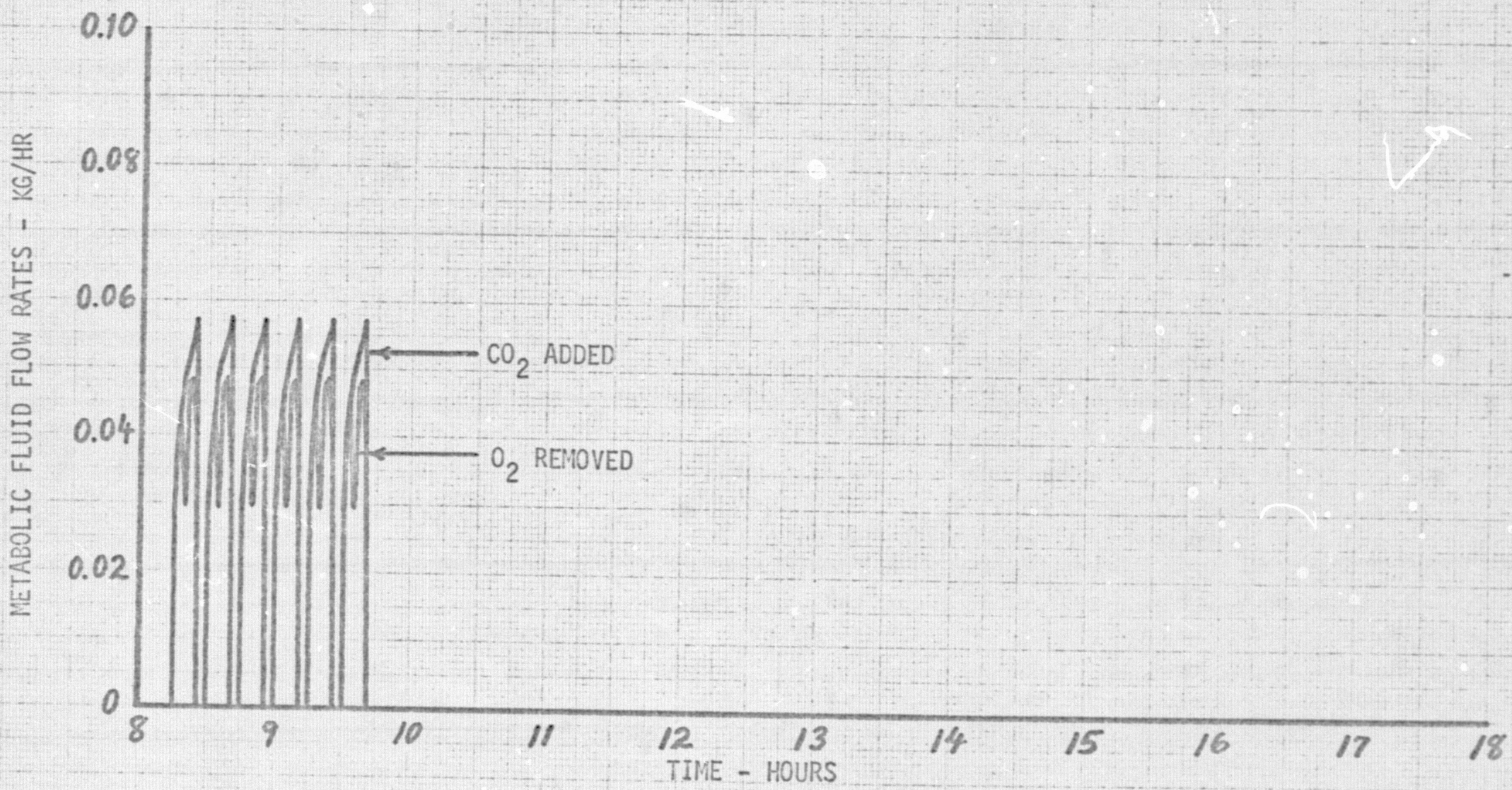

Figure 6-15. Net CO_2 Addition and O_2 Removal Rates from Space Station Shower Loop

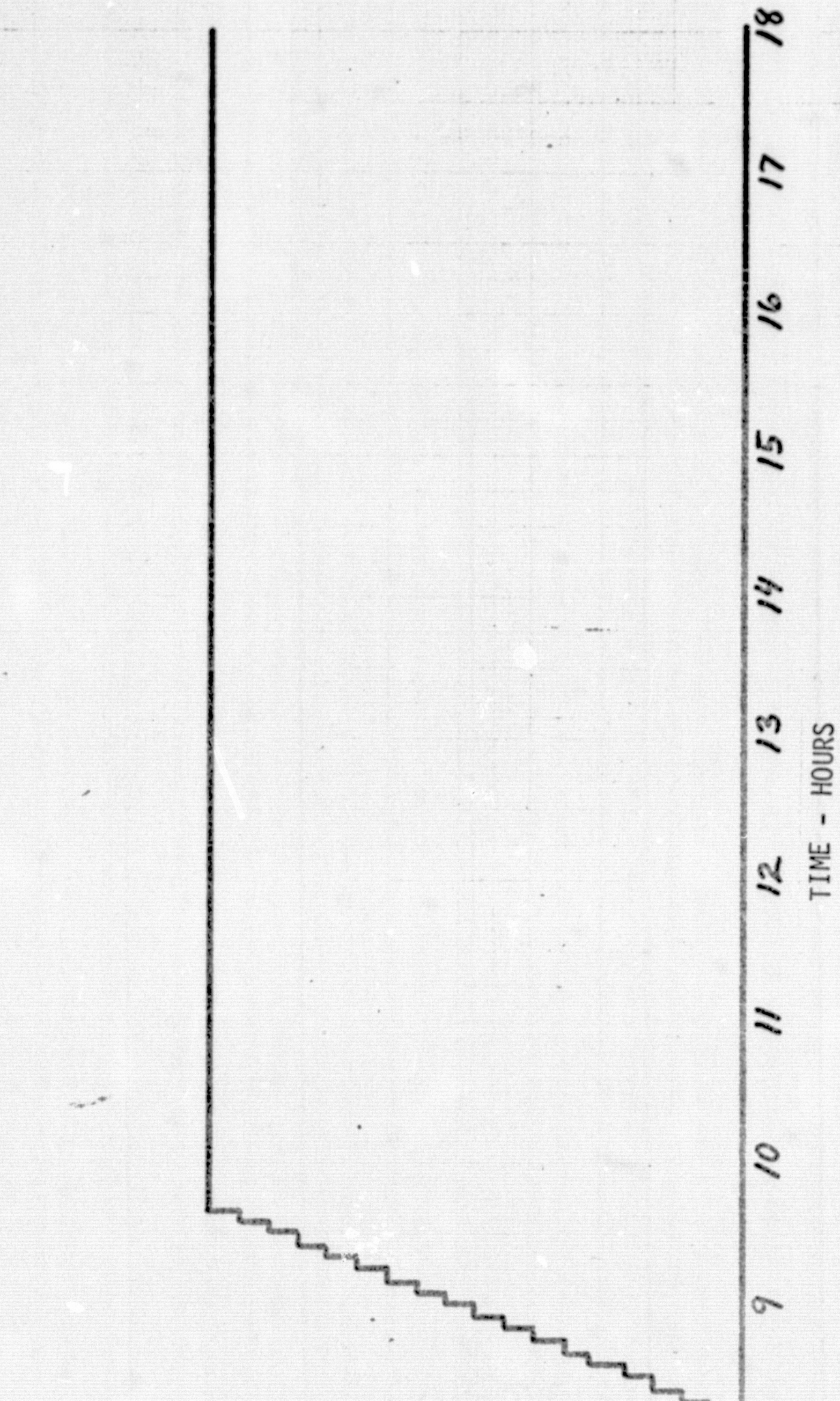

Figure 6-16. Space Station Water Usage by Shower

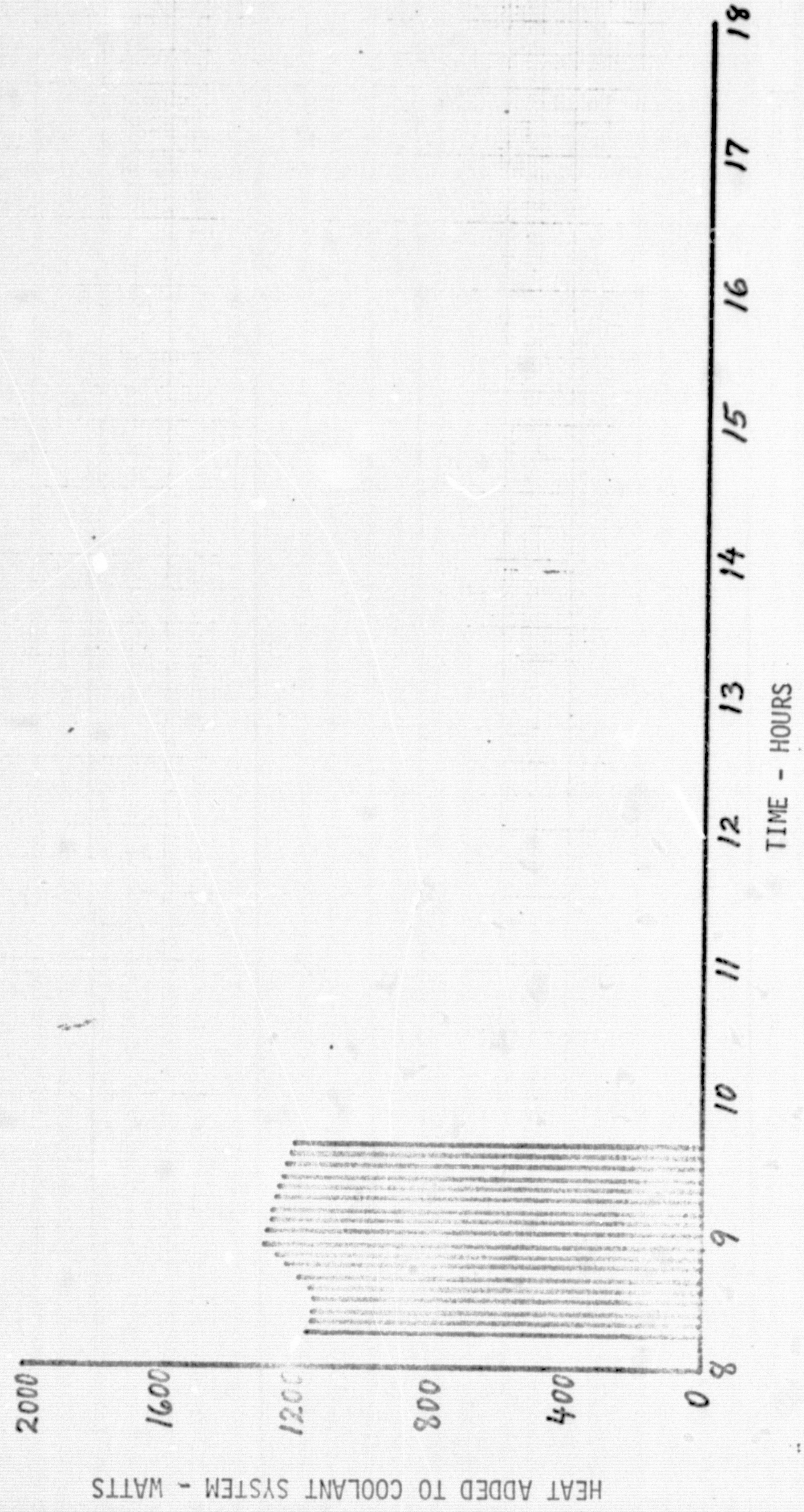

Figure 6-17. Heat Input to Space Station Coolant from Shower

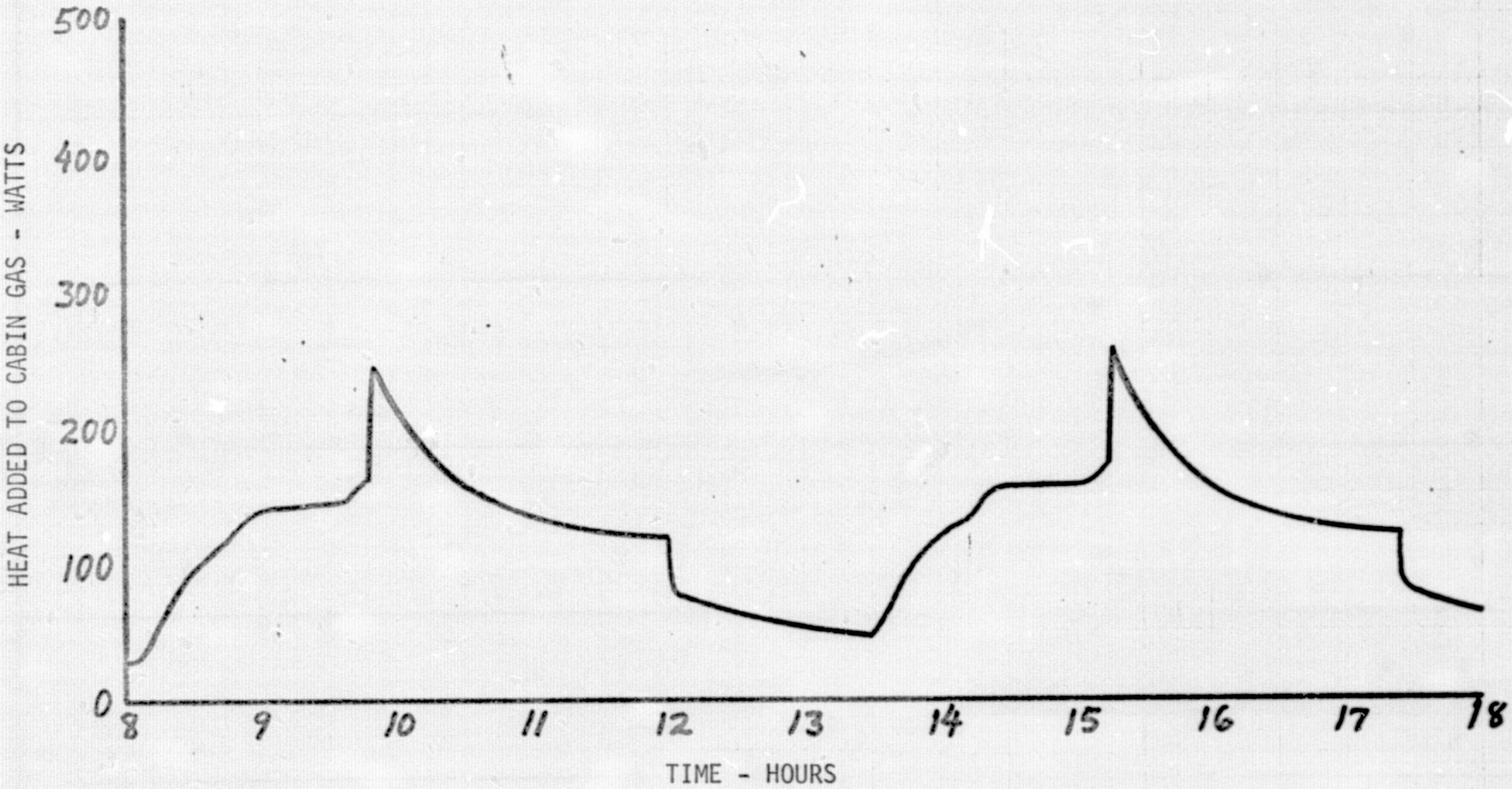

Figure 6-18. Space Station Sensible Heat Input from Clothes Washer/Dryer

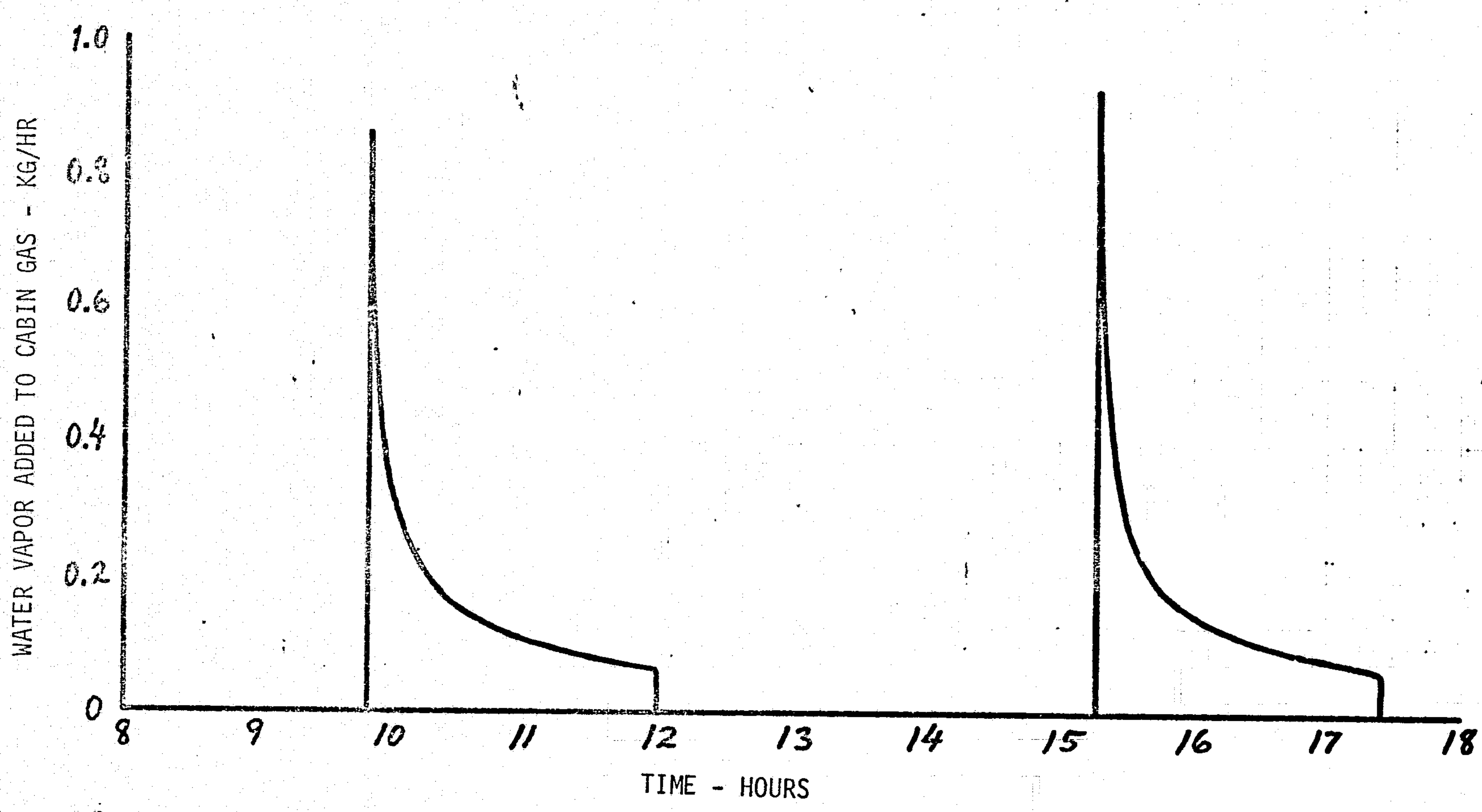

Figure 6-19. Space Station Water Vapor Input from Clothes Washer/Dryer

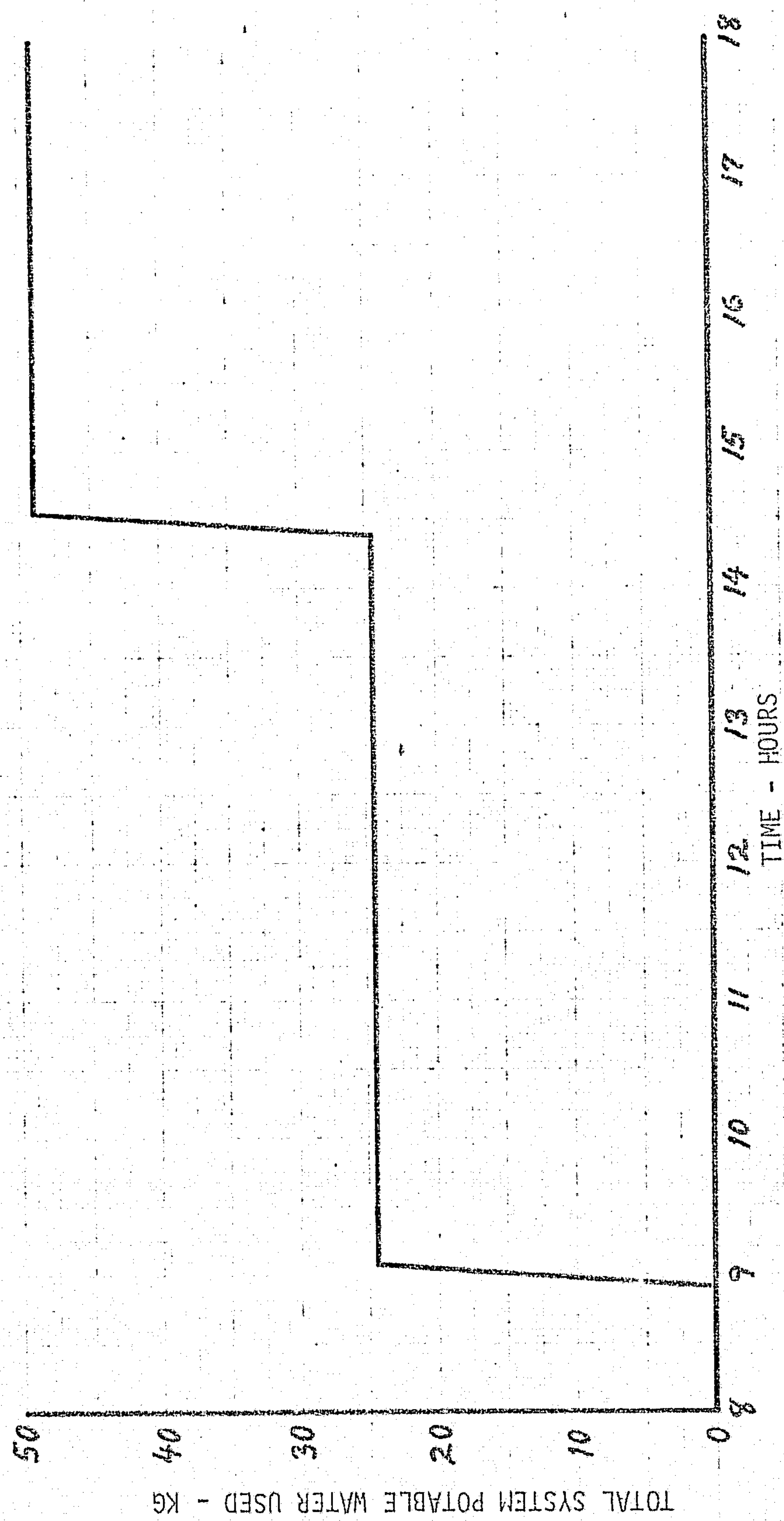

Figure 6-20. Space Station Water Usage by Clothes Washer

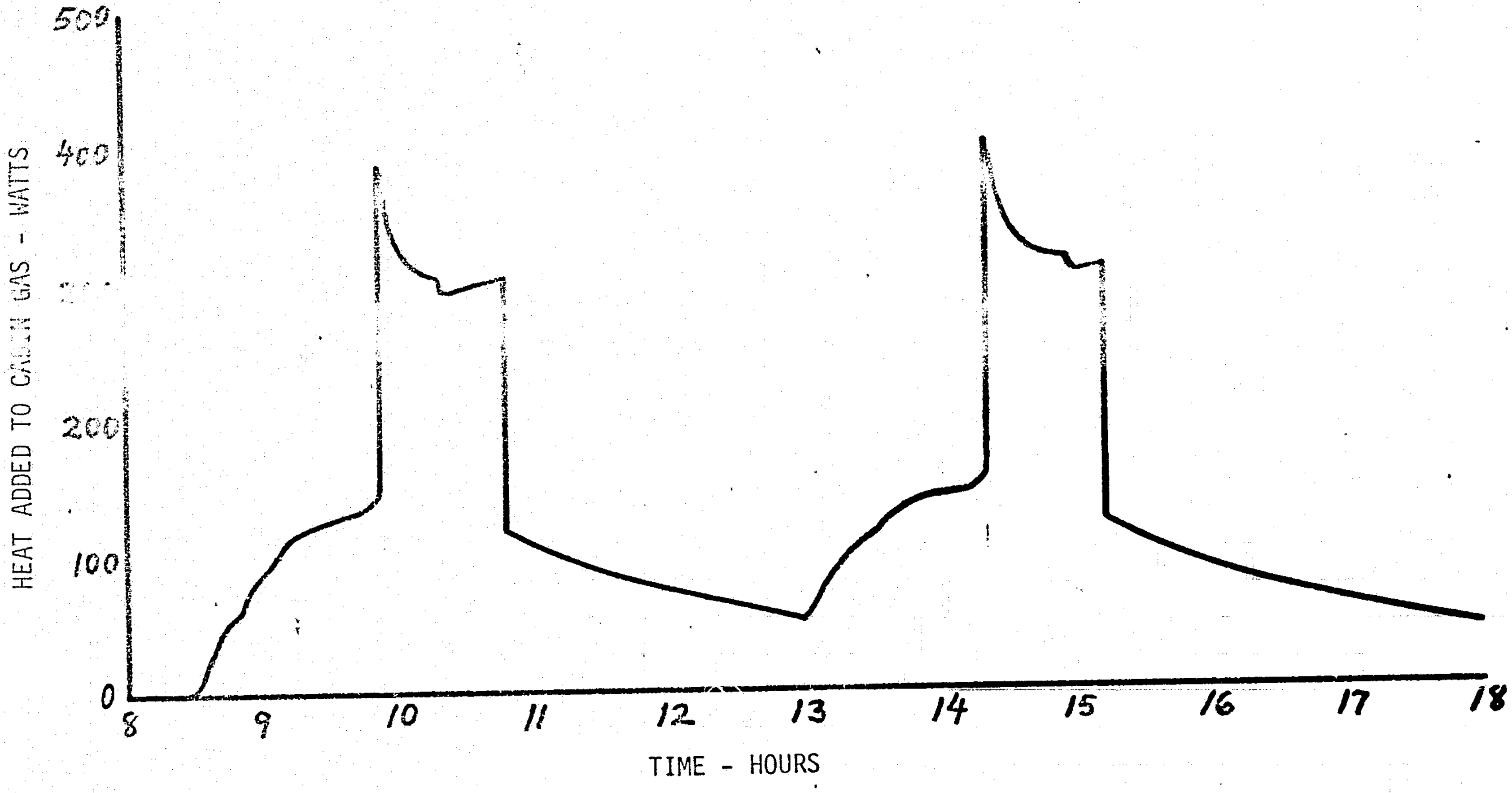

Figure 6-21. Space Station Sensible Heat Input from Dishwasher/Dryer

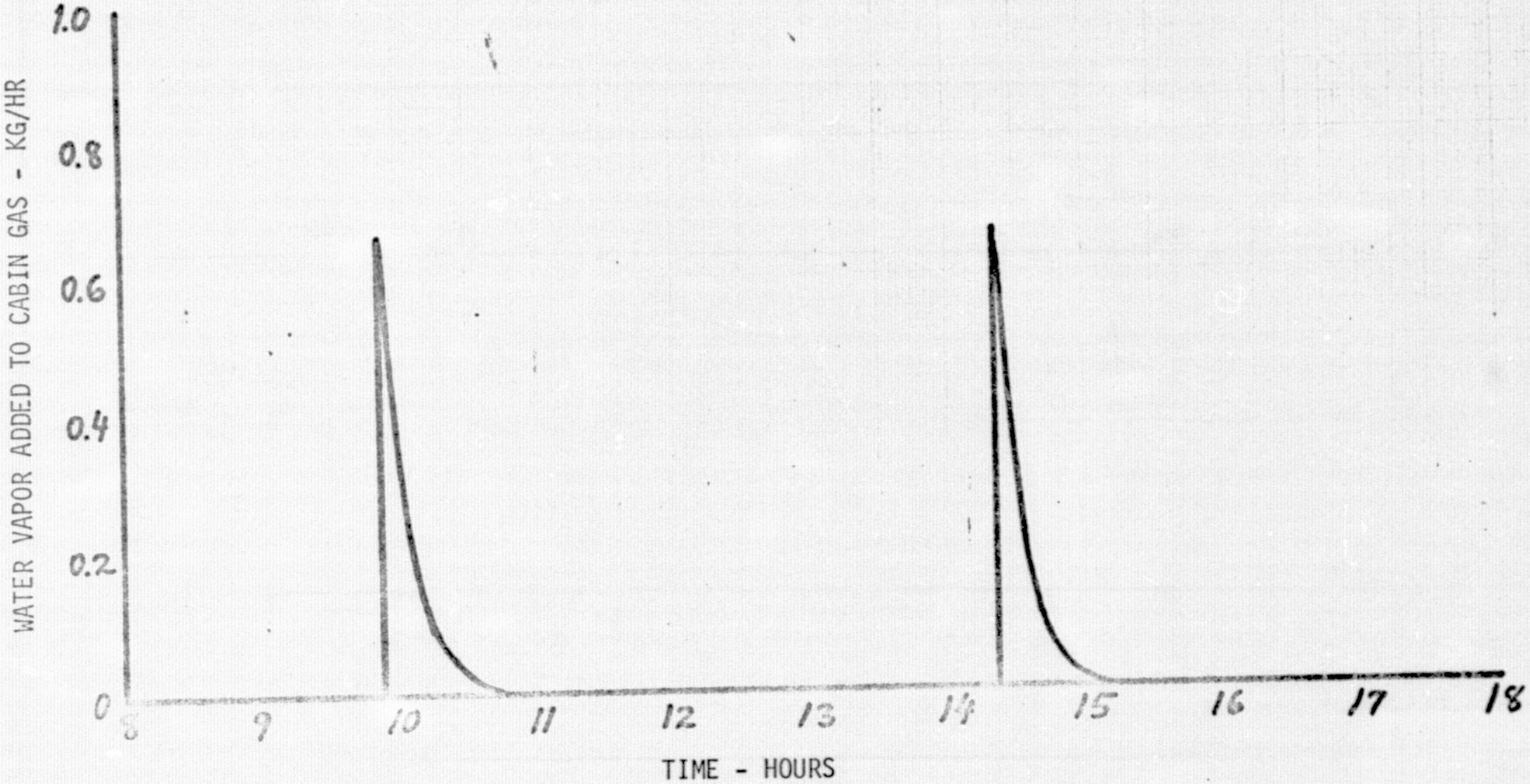

Figure 6-22. Space Station Water Vapor Input from Dishwasher/Dryer

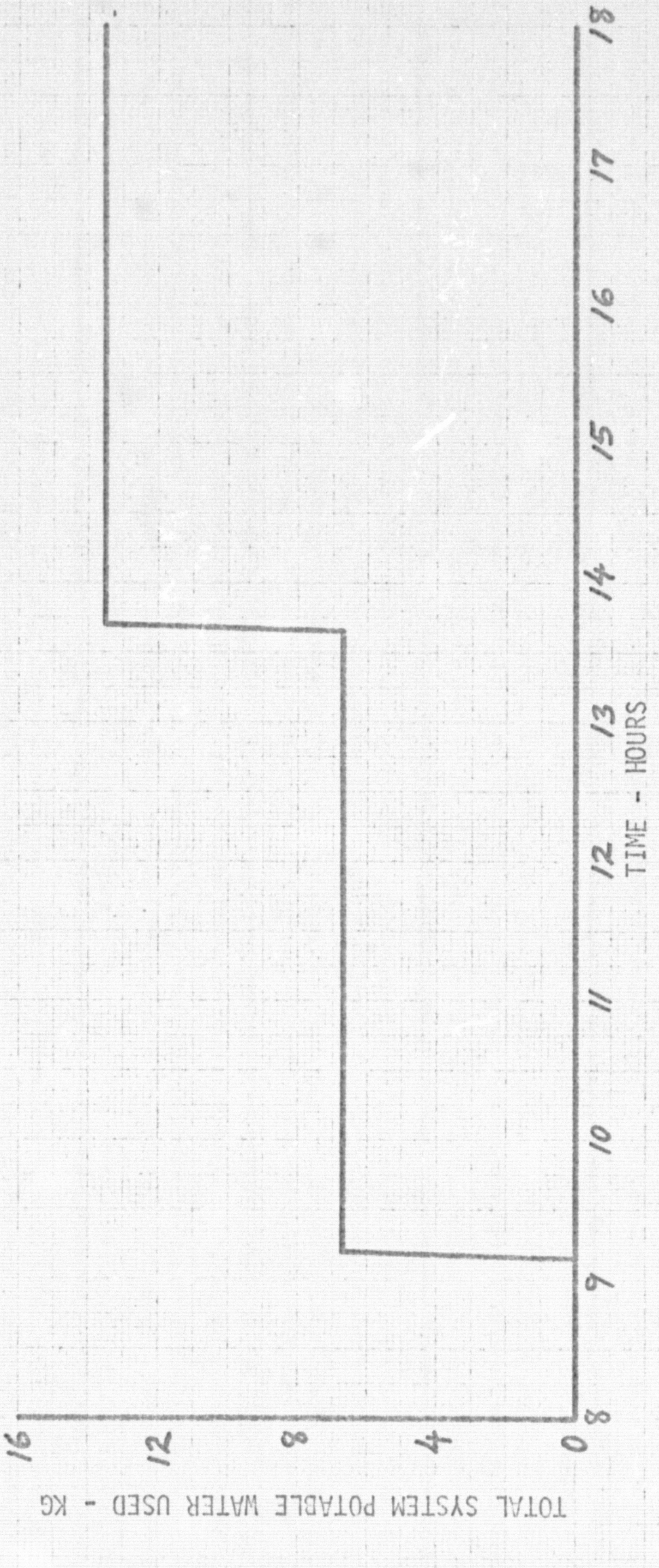

Figure 6-23. Space Station Water Usage by Dishwasher

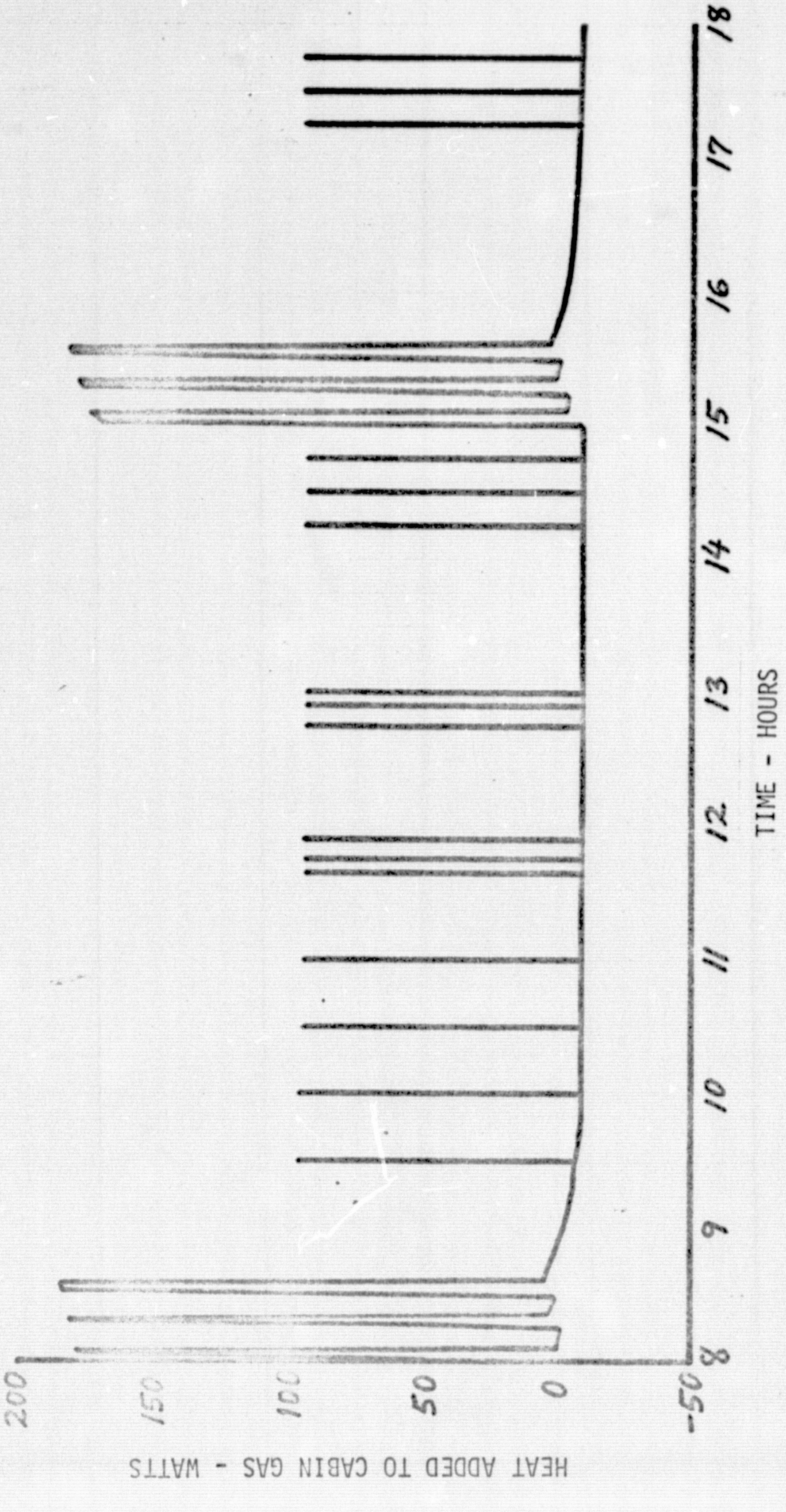

Figure 6-24. Space Station Sensible Heat Input from Dryjohn

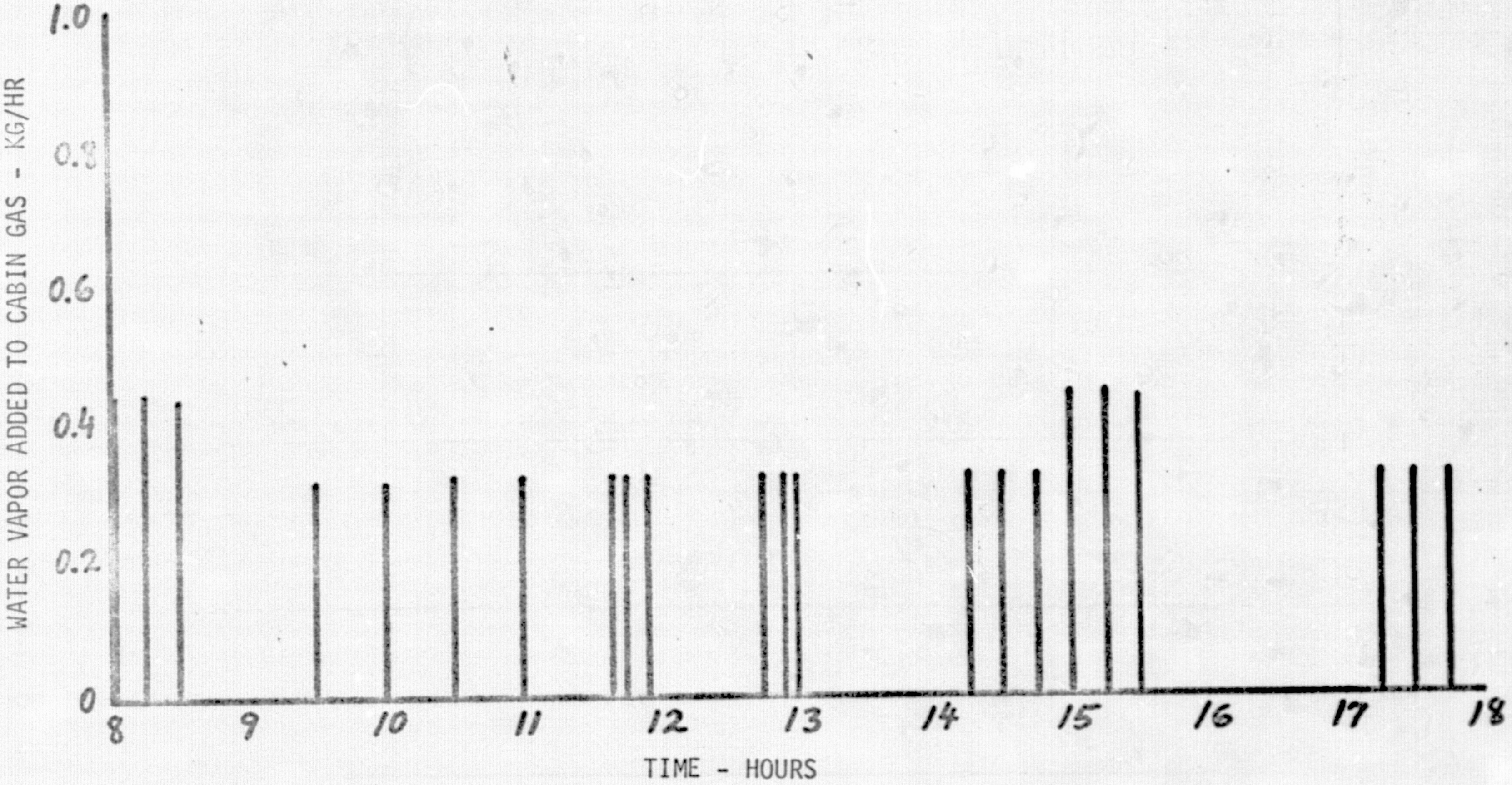

Figure 6-25. Space Station Water Vapor Input from Dryjohn

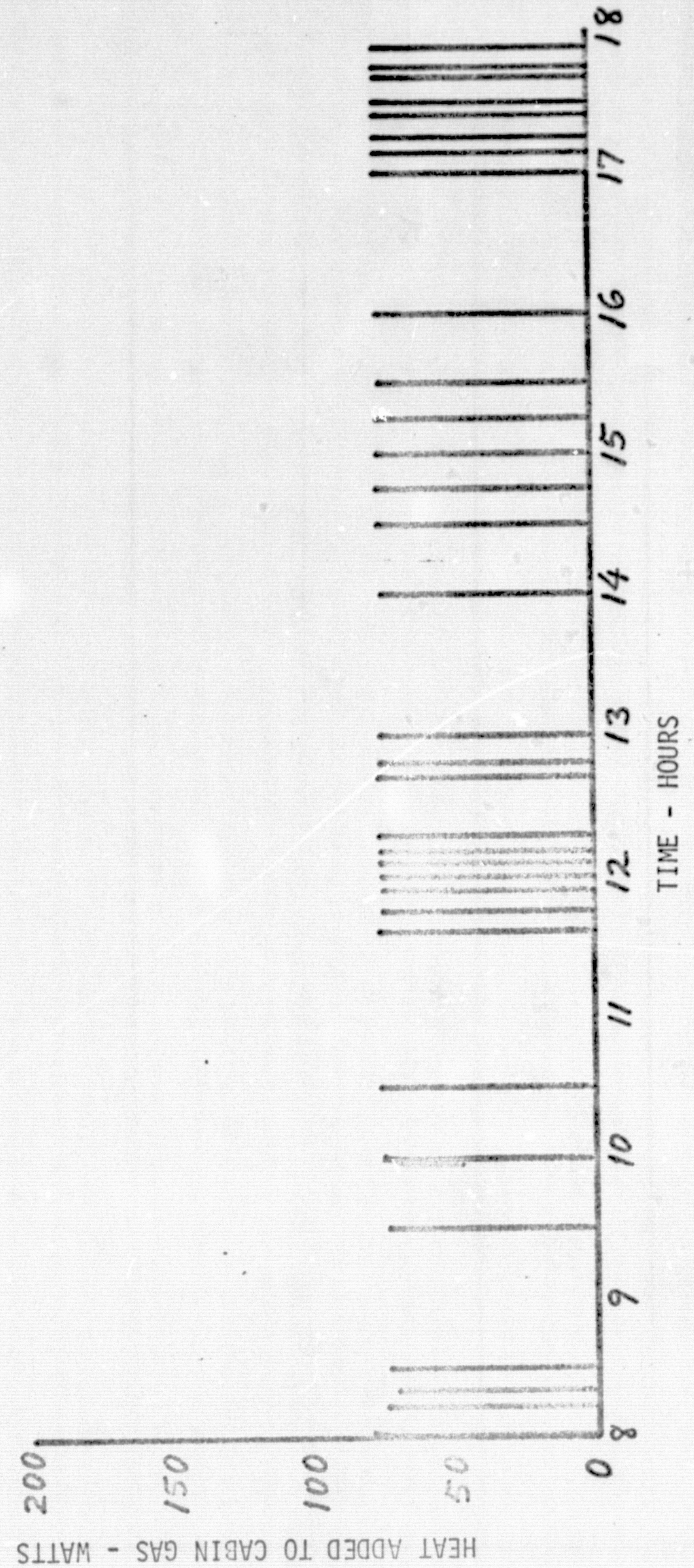

Figure 6-26. Space Station Sensible Heat Input from Wet Wipe Wetter

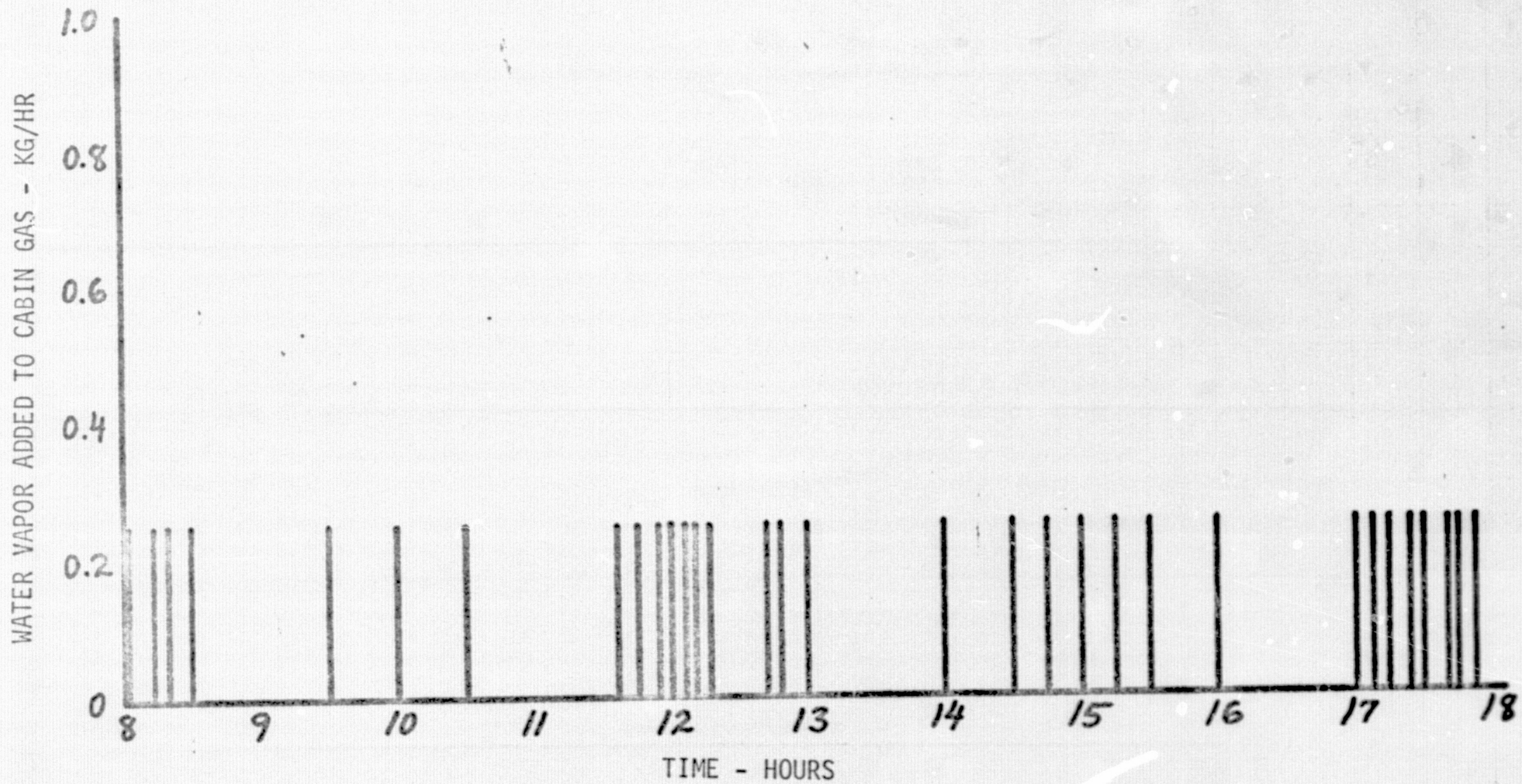

Figure 6-27. Space Station Water Vapor Input from Wet Wipe Wetter

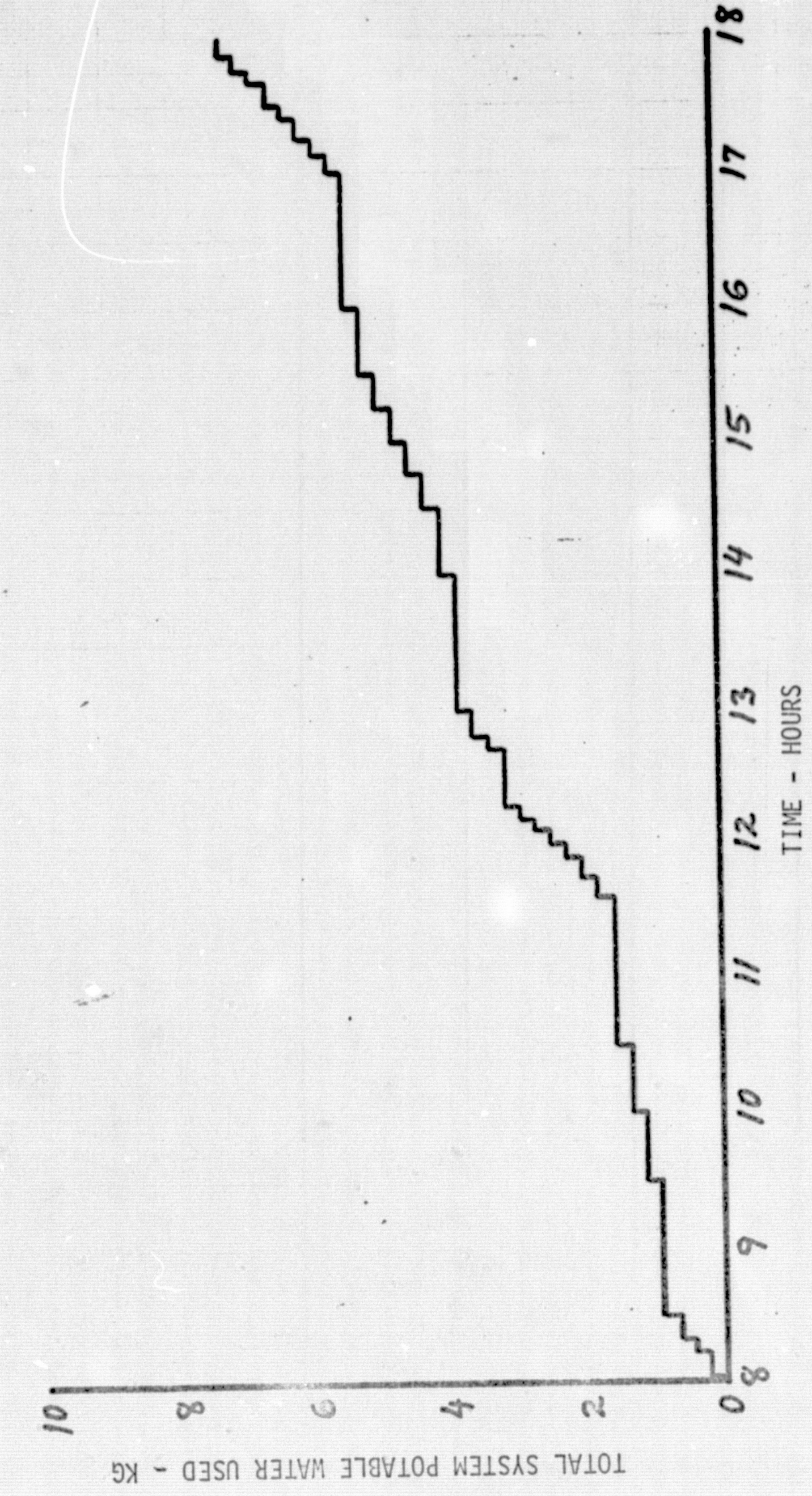

Figure 6-28. Space Station Water Usage by Wet Wipe Wetter

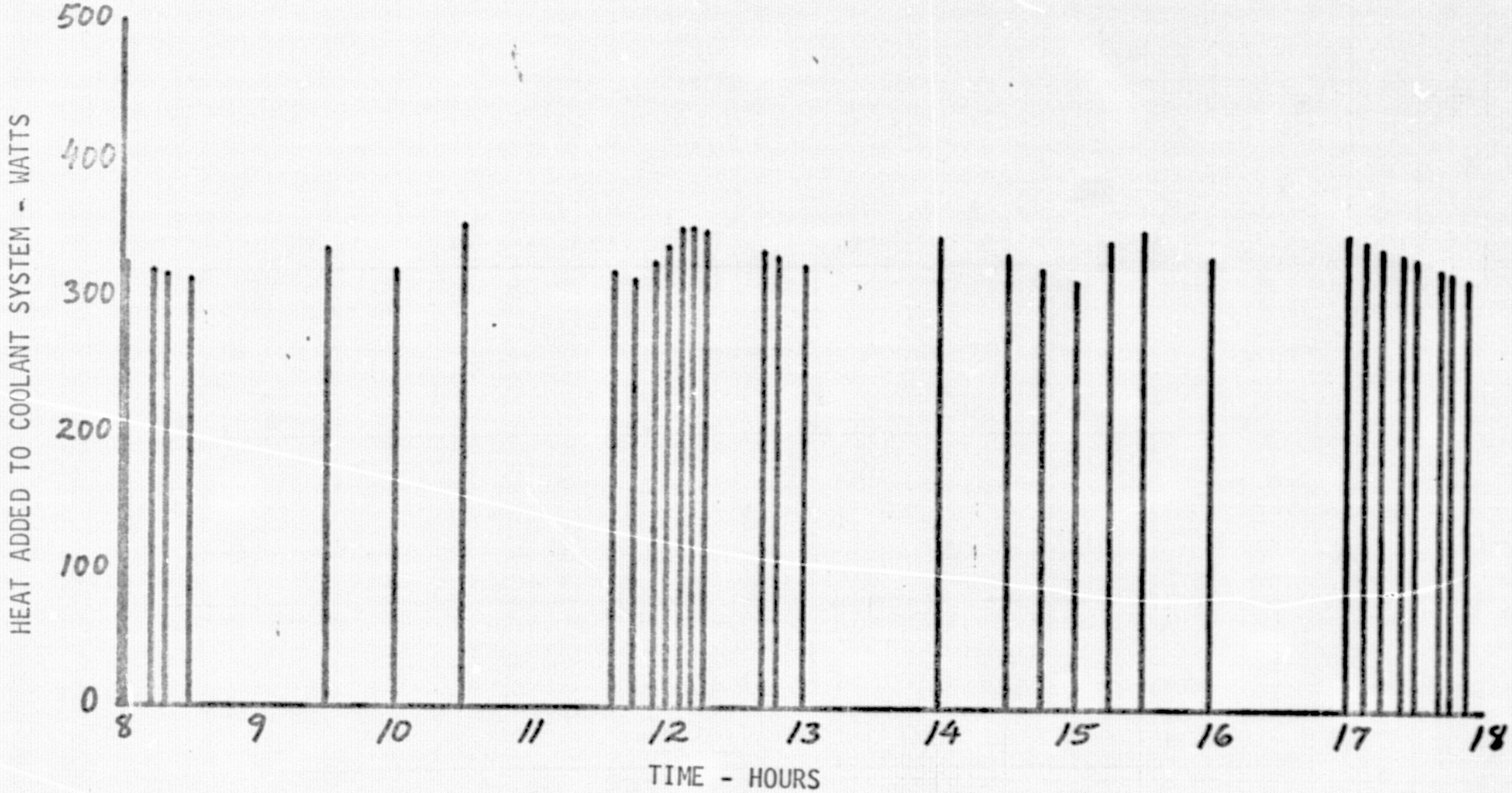

Figure 6-29. Heat Input to Space Station Coolant from Wet Wipe Wetter

7.0 CONCLUSIONS AND RECOMMENDATIONS

New G-189A subroutines have been developed to model future spacecraft crew appliances. These subroutines, some of which model more than one component, are generally described as follows:

CHILLR (simulates a thermally insulated locker cooled either by an externally chilled fluid or a self-contained refrigeration unit)

* Freezer
* Refrigerator

FTRAY

* Food warming/serving tray (Skylab-type)

ROSMOS

* Reverse osmosis waste water treatment unit

SHOWER

* Spacecraft whole body shower

WASDRY

* Clothes washer
* Clothes dryer
* Combined clothes washer/dryer
* Dishwasher/dryer
* Towel/cloth drying rack

WASTEC

* Dryjohn
* Urinal

The new appliance subroutines have been described, with user's instructions and verification and demonstration results presented for each. The appliance models have been shown to be valid and operational in an all-up G-189A system model. Most of the crew appliances described in this report have not yet been built; several have not been

7.0 (Continued)

designed for zero-g application. Consequently, the verification of the accuracy of the models was limited largely to independent analysis. It is therefore recommended that the new appliance subroutines be closely correlated with future experimental data when testing of the actual appliance hardware is done.

Several of the crew appliances which have been modeled are relatively complex subsystems involving a number of components such as pumps, fans, accumulator tanks, heaters, valves, etc. In such cases, new subroutines have been developed only for the individual components which were not already available in the G-189A subroutine library. Consequently, a number of G-189A components may be required to simulate a single appliance (e.g., 12 components were required to model the clothes washer/dryer). Appliances such as these could be modeled and run by themselves using a typical range of ambient conditions such as temperature and humidity. The results from these runs could be converted into tabular data which would be used in a new subroutine to simulate the performance of the entire appliance subsystem. This new subroutine would accept, as input data, the actual conditions existing during a G-189A run. It would then determine from the tables the required performance parameters (i.e., temperature rise, humidity change, and heat dissipated to the cabin) for the appliance subsystem. This effort would allow much simpler modeling and faster computer execution of large ECLSS's involving many appliances.